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"Reliable, Economic, Efficient CO2 Heat Pump Water Heaters for North America"

Submitted by

United Technologies Research Center

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ABSTRACT

Adoption of heat pump water heating technology for commercial hot water could save up to 0.4 quads of energy and 5 million metric tons of CO_2 production annually in North America, but industry perception is that this technology does not offer adequate performance or reliability and comes at too high of a cost. Development and demonstration of a CO_2 heat pump water heater is proposed to reduce these barriers to adoption. Three major themes are addressed: market analysis to understand barriers to adoption, use of advanced reliability models to design optimum qualification test plans, and field testing of two phases of water heater prototypes. Market experts claim that beyond good performance, market adoption requires "drop and forget" system reliability and a six month payback of first costs. Performance, reliability and cost targets are determined and reliability models are developed to evaluate the minimum testing required to meet reliability targets. Three phase 1 prototypes are designed and installed in the field. Based on results from these trials a product specification is developed and a second phase of five field trial units are built and installed. These eight units accumulate 11 unit-years of service including 15,650 hours and 25,242 cycles of compressor operation. Performance targets can be met. An availability of 60% is achieved and the capability to achieve >90% is demonstrated, but overall reliability is below target, with an average of 3.6 failures/unit-year on the phase 2 demonstration. Most reliability issues are shown to be common to new HVAC products, giving high confidence in mature product reliability, but the need for further work to minimize leaks and ensure reliability of the electronic expansion valve is clear. First cost is projected to be above target, leading to an expectation of 8-24 month payback when substituted for an electric water heater. Despite not meeting all targets, arguments are made that an industry leader could sufficiently develop this technology to impact the water heater market in the near term.

TABLE OF CONTENTS

EXECUTIVE SUMMARY

Heat pump water heaters can provide hot water at higher efficiency than electric or gas water heaters. Adoption of this technology for commercial hot water could provide energy and CO2 emission savings of up to 0.4 quads and 5 million metric tons annually. However, market acceptance of this technology has been limited by perceptions of limited operating range, poor reliability and high cost. The objective of this project is to demonstrate that a commercial 60 kW CO2 HPWH can be designed and fielded with sufficient performance and reliability to overcome these perceived barriers.

The scope of this project covered three themes. First, market experts were surveyed to understand exactly what would be required to gain market acceptance of heat pump water heater technology in commercial application. These requirements were then used to generate performance, reliability, certification, and cost targets for a production device. To offset the expense of qualification testing, advanced reliability models were developed to optimize component and system qualification testing requirements to meet the proposed reliability target. Finally, to prove the concept, two phases of heat pump water heaters were designed, fabricated, field tested, and destructively examined to demonstrate that performance, reliability, and cost targets could be met.

Market studies highlighted three major issues. It was noted that the incremental cost of generating hot water was small but the cost of replacing that hot water in the event of a failure was very high. For this reason, market experts said reliability was the foremost concern. A “drop and forget” system was needed to achieve market acceptance. A maximum first year failure rate target of 7.6% was proposed based on Carrier marketing experience. The second issue is the time period required to recoup installed cost, a metric that addresses both cost and performance. Experts indicated that a payback period of six months or less was required for acceptance in new construction, although that requirement might be relaxed somewhat if the unit was offered by a recognized brand name. To give the highest possible performance over the widest range of operating conditions supercritical cycle CO2 heat pump water heater technology was chosen. This technology promises an average coefficient of performance (COP) of 3.6 in North America while providing good efficiency in ambient temperatures as low as -20 ºC. Given this performance target a first cost target was developed to give a six month payback at 2003 electricity costs.

Carrier has been developing a CO2 heat pump water heater system for the European market for several years. This system, which continued development in parallel to the North American project, provided a strongly leveraged basis for a North American water heater design. Three phase 1 North American units, modified from the European prototypes as little as possible to accommodate electric supply and safety code restrictions, were fabricated in a Carrier factory and installed in the field. Industrial process, food service, and laundry sites in the Southeast and Northeast were chosen for this phase of trials. The units were operated 6 unit-months and then lessons learned were applied to produce a North American product specification. Six units were built to this specification, with five units fielded and one unit maintained in a test room at UTRC for development and troubleshooting. A Northwest site was added in this phase, and applications were expanded to include general hot water in a college gymnasium, a restaurant, a hospital, a hotel and also process water in a food processing facility. These trials accumulated a total of 11 unit-years of service, including 17,650 hours and 25,000 cycles of compressor operation. Performance and reliability were tracked remotely. Once complete, these field units were retrieved and subject to destructive examination to uncover incipient problems and determine rates of component wear.

The field trial units exhibited an average COP of approximately 2.8 at the design condition of 10 ºC ambient and 60 ºC water delivery temperatures, retaining a COP of 1.8 at -10 ºC. Unit performances were 20-30% below the target value. This performance was comparable to contemporary European units but much lower than early European units. The deficit was proven to result from poor compliance with compressor valve plate flatness specifications. With good quality control the design performance could be met or even exceeded.

The high system pressures and advanced controls needed to achieve this performance can affect reliability. Reliability targets were not met, with an average of 3.6 failures per unit-year in the phase 2 trials. Although this figure is far above the target, most of the observed control, sensor, and wiring failures are very similar to other new HVAC development, thus giving high confidence that these failures can eventually be reduced to target levels. The disproportionate number of leaks observed can be addressed through specific measures, leaving the electronic expansion valve as the one item that requires further development in order to project a path to the reliability target. Root cause analysis on failures of this component suggested that only development, not invention, is required to achieve the target. The compressor was observed to behave similarly to European units, with good reliability but relatively short predicted lifetime.

Availability of the units was found to average 60% over the term of the trials, with a high of 84% and low of 44%. These values are lower than would be experienced by a real product because UTRC personnel were not able to maintain or repair units as promptly as Carrier Commercial Services would. If two days were allowed for each repair and the excess delay in servicing removed from the availability calculations, average availability rises to the low 90% range.

Projected costs at quantity were found to be 20% higher than the target value, resulting in payback intervals of 8-24 months in different regions. Short compressor life, ancillary systems required to ensure even this relatively short compressor lifetime, and a double-walled heat exchanger for sanitary water use were primarily responsible for the increased cost.

Although not all targets were met it appears that performance and reliability targets can be met at a cost somewhat higher than the target cost given a reasonable additional development and qualification period. The primary risk to market acceptance then becomes somewhat longer payback periods than those dictated by market experts. Two factors could act to reduce that risk. First is the market impact of an industry leading supplier of the system. Second is the direct impact of rising electricity costs on payback. Given these unknown factors, there is a good probability that near term development of CO2 heat pump water heater technology could result in significant market impact.

RESULTS AND DISCUSSION

Objectives of the Project

The primary purpose of this project was to lower the primary barriers to acceptance of highly efficient heat pump water heater (HPWH) technology in the US market. The barriers were proposed to be performance, reliability, and first cost. These barriers could be real or perceptual.

Performance was addressed through the innovative use of carbon dioxide (CO2) as a refrigerant. The relatively low critical temperature of CO2 (31 °C) was utilized to develop systems with higher efficiency (up to 4 times that of gas or electric systems), higher water delivery temperature (up to 80 °C), and wider operating range (ambient temperatures as low as –20 °C) than current heat pump systems. Reliability was addressed through a coordinated effort of modeling and simulation to guide the design and execution of testing, prototype development, and the deployment of eight field trial units. Special focus was given to the compressor, charge leakage, and the electronic expansion valve (EXV), because of their historical reliability issues. We also engaged the Carrier Commercial Service organization to begin developing nationwide installation and service capability to help address perceived reliability. We propose that first cost can be addressed by leveraging UTC Carrier's global marketing and sourcing network to establish economies of scale. Toward this goal we engaged Carrier Building System and Services Lead Design Centers to develop quality suppliers and begin evaluation of the North American market. The improved performance of the CO2 HPWH system, supported by the Carrier service organization, is expected to drive down operating costs and partially ameliorate the first cost issue.

The project was divided into 14 tasks, but the main flow of work falls into three themes that primarily address reliability and performance: reliability modeling and component qualification testing; market requirements and development of specifications; and three iterations of system design supported by two phases of field trials.

CO2 heat pump water heaters are a new market for Carrier, and this market is clearly affected by perceived reliability of the product. Field trials are required to ensure reliability, but these trials can be difficult and expensive to field for innovative new technologies. We developed advanced reliability modeling to reduce the overall quantity of test units and testing hours required to demonstrate reliability. The modeling was intended to validate and possibly improve the requirements for component qualification testing, to evaluate the system reliability that can best be expected from the proposed field trials, and finally to track the evolution of estimated reliability as data was derived from the field trials.

To develop technology that overcomes barriers one must first identify and quantify the barriers. Entering this project we identified the barriers as reliability, cost, and performance. Interviews with market experts were performed to validate the accuracy and completeness of this set of barriers and to quantify metrics. These experts also indicated requirements for certifications that would be necessary or desirable in the US market. This information, along with the component testing requirements produced in the first theme, was then used to develop detailed specifications for the system. Along with requirements and specification development, a tool for specifying the size, number, and cost of units for a commercial installation quote was initiated.

Early in the project an initial system specification was produced from incomplete requirements to facilitate production of a first round of prototypes. These prototypes were based very closely on the HPWH system Carrier has been developing in Europe. Three systems were produced, tested at UTRC and then installed in the field. In parallel to these field trials, development of qualification protocols and a more carefully researched specification was pursued. The phase 1 trials thus provided very early feedback as to any problem areas of the system, allowing us to address issues before designing and fielding the phase 2 prototypes. The phase 1 trials accumulated approximately 6 unit-months of field operation before the final specifications were completed. Six systems of the final specification were produced with five units fielded and one unit retained in a UTRC lab to simulate any field failures. The phase 2 units accrued 62 unit-months of operation. Combined with the phase 1 units a total of 132 unit-months of site experience were accumulated. All eight fielded units were tracked for component and control failures, and ultimately torn down and inspected for wear and other deterioration to validate the reliability models. Finally, all data was applied to develop a final product specification.

CO2 HPWH Technology

Heat pump water heaters have been in commercial use since the 1960s. The concept is straightforward; cold refrigerant is first boiled by extracting heat from the atmosphere, then the resulting vapor is subsequently compressed to a high temperature, cooled by heat transfer to the water to be heated, and finally isenthalpically expanded to the original cold liquid state. Figure 1.1 illustrates a generic heat pump water heater.

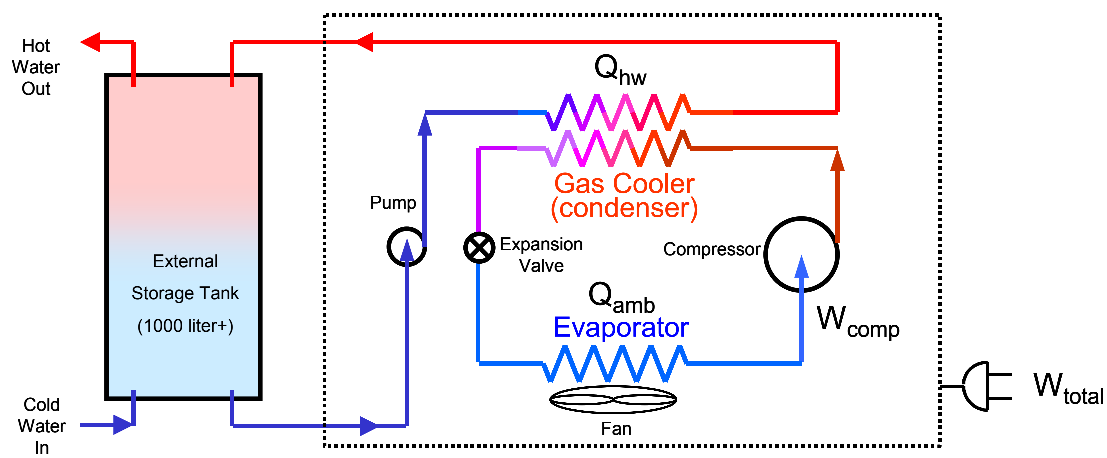


Figure 1.1: Schematic of a heat pump water heating system

CO2 HPWH technology has the potential to deliver the benefits of both conventional HPWHs and electric or natural gas water heaters: high efficiency, high reliability, high water delivery temperature, and year-round operation. The performance advantage of CO2 is illustrated in Figure 1.2 by comparing vapor compression cycles using R-134a and CO2 (R-744), given typical cycle temperatures for water heating. CO2 has a relatively low critical temperature of 88 °F (31°C), so, unlike R-134a, the CO2 cycle becomes transcritical and heat rejection occurs at temperatures above the two-phase region. Since there is no phase transition during heat rejection, continuous temperature glide throughout the gas cooler (comparable to the condenser in a two-phase system) results. This continuous temperature glide provides a better temperature match for heat transfer to the circulating water than in condensing systems, resulting in improved heat exchanger effectiveness and higher water delivery temperatures given a comparable area for heat transfer. Specifically, temperature glide and high gas cooler effectiveness mean that high water delivery temperatures (up to 80 °C) are possible without a significant decrease in system efficiency. In fact, efficiency can improve because better heat transfer in the gas cooler reduces the compressor load, as the compressor does not need to operate at high pressure ratios to generate this high discharge temperature for effective water heating. Based on UTRC modeling and testing of full-scale laboratory prototype HPWHs, a steady state annual average COP of 3.73 is possible across the continental US for 140 °F (60 °C) water delivery temperature while A COP of 3.23 is achievable for 176 °F (80 °C) water delivery temperature.

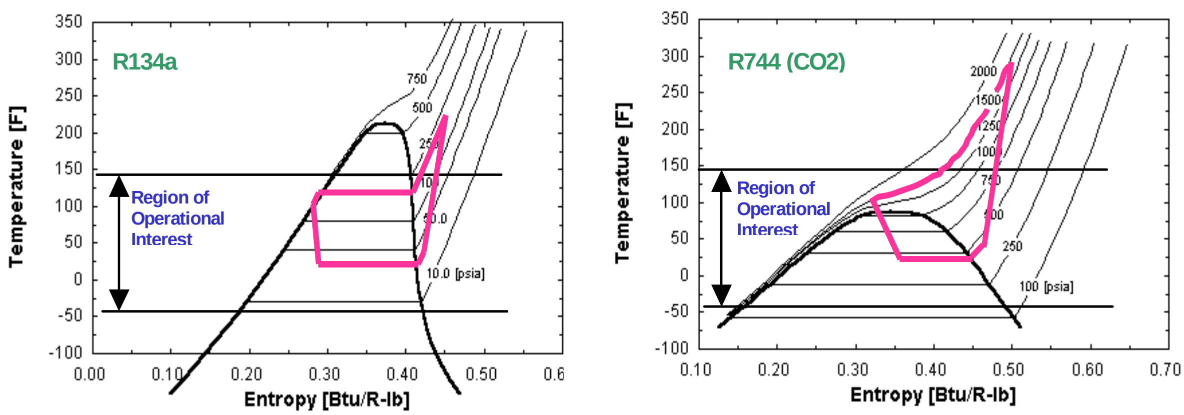


Figure 1.2: T-S diagrams of conventional (saturated) and supercritical heat pump cycles

As with traditional HPWH technology, the CO2 HPWH system capacity varies with ambient temperature. However, it is less sensitive to ambient temperature swings and allows good efficiency and high delivery temperatures at much lower ambient temperatures than existing hydrofluorocarbon (HFC) based systems. The main reason for this is the relatively high evaporating pressure for CO2 systems, which results in lower compression pressure ratios and thus less sensitivity to varying operating conditions. Also, high pressure CO2 has excellent thermal properties when compared to conventional HFC refrigerants (R-134a, R-407C, and R-410A), such as higher thermal conductivity, higher specific heat capacity, and lower kinematic viscosity, allowing for higher heating capacity per unit mass of refrigerant and improved heat transfer. Overall, these properties result in a smaller size unit, with the potential for significantly reduced footprint and manufacturing cost.

Finally, beyond the performance benefit, there is an additional environmental benefit to using CO2 rather than HFCs. CO2 is non-toxic, non-flammable, has an ODP of zero and a GWP of 1. Even the most advanced 'environmentally friendly' HFC refrigerant, R-410A, has a GWP that is roughly 1700 times higher over an integrated time horizon of 20 years [HPP-AN22-4]. Because of this there are no refrigerant recovery requirements for CO2, which makes it an attractive option from an installation and service point of view.

Task-by-Task Accomplishments

Task 2 – Reliability Modeling

Much of the description of the reliability methodology presented here has been previously published by UTRC as noted in the bibliography.

Reliability is perhaps our most critical barrier to overcome. To predict reliability of the heat pump water heater some reliability data must be generated. Because this is a relatively high risk market the investment, and therefore the number of CO2 HPWH prototypes to be produced and tested, would necessarily be limited. The team has developed a methodology for reliability prediction of new products where field data may not be available, the allowed number and length of experiments is limited, and the majority of reliability tests are right censored.

A typical industrial approach to ensuring reliability for a product relies on qualifying the product after observing no (or possibly very few) failures over a regimen of testing over the design envelope. Such tests are referred to as qualification tests. In one class of qualification tests, the main objective is to meet several predefined targets for system operations under extreme conditions where the targets are largely derived from experience (i.e. the tests are typically not used to derive quantitative life predictions). In another class of tests, several systems are run in parallel to collectively span a large number of operating cycles in a relatively short period of time. In both test classes the number of operating cycles is accelerated but the operating conditions are not. By observing no (or possibly a small number of) failures during such qualification tests, quantitative estimates of field failure rates at desired confidence levels are derived using standard statistical formulas.

Accelerated life testing (ALT) has been recognized as a methodology that can usefully predict service life with less time commitment than traditional qualification testing. Since service life of a product is much longer than the product testing and qualification cycle, ALT proposes testing under conditions that are designed to accelerate failures. The underpinning premise is that service life can be extrapolated from accelerated tests with use of models usually referred to as acceleration models. For this reason, ALT usually requires rich data sets consisting of many observations and observed failures. Combining acceleration models with probability distributions of time to failure provides a way to infer field reliability from test data analysis. A number of models and methods are commonly applied. Applied to a product like the CO2 HPWH, accelerated testing would typically be applied to straightforward tests where the acceleration factors are well understood such as exterior corrosion testing or valve seal wear. It is more difficult to apply in cases where there are multiple potential failure modes, each with a different acceleration factor, especially on new technology where the dominant failure mode may not even be identified.

Methodology

At a high level then, the problem of interest is estimation of unknown reliability parameters of a given system at component and/or system levels utilizing experimental and possibly field data. Of special interest are cases where data is limited, e.g. when field data is sparse/non-existent and experimentation is expensive. Here we propose a methodology for simultaneous design of experiments and estimation of bounds on reliability parameters such as service life or failure rates at desired confidence levels. Development of bounds on reliability parameters is proposed because of inaccuracy of reliability predictions in sparse data environments. Moreover, development of the tightest bounds on the reliability parameters is proposed as the criterion for design of experiments. The problem is formulated as an optimization, making it possible to make explicit tradeoffs between the cost and benefit of additional experiments.

Similar to ALT, the methodology presented here allows reliability estimation by extrapolation from experiments. Unlike ALT, any use of acceleration models is limited to bounding mission reliability at a desired confidence level (as opposed to accurately predicting it). Indeed, we consider a limit case where analytical expressions for optimal experiments are derived with no information about failure acceleration. This is advantageous since in many situations, system complexity, existence of many failure modes, and expensive tests prohibit extensive experiments to obtain rich data required for making accurate predictions. When no information is available, the methodology can produce mission reliability predictions commonly used in industrial applications. Where better information is available, the methodology allows incorporation of models for failure propagation, failure acceleration, and system operations to develop predictions at higher levels of accuracy.

For the purpose of this presentation, we consider the first year field failure rate (FFR) as the reliability metric of interest. However, the methodology is generically applicable if other measures such as mean time between failures are of interest. We define FFR as the probability of at least one failure during product's first year in service, where a "failure" is defined as any event that entitles the product owner to a warranty claim. Likewise, survival within a period of interest means not observing any warranty claims during that period.

The developed methodology relies on three steps: estimating a set where the unknown parameters are most likely to be found, calculation of an upper bound for the reliability metric of interest conditioned that the parameters reside in the estimated set, and finally tightening the bounds via design of experiments. Various forms of information can be incorporated to arrive at a better prediction of FFR. Failure mode decomposition, fault tree analysis, acceleration models, expert judgment, and prior history are all examples of this. By decomposing the overall system failure into failure modes that affect component or part levels and applying more accuracy in either models, data, or even informed judgment, we will be able to enhance the accuracy of the predictions.

To begin the process we first identify for each of the components of interest all the operation modes whose duration or number of relevant cycles may be correlated to system failure. We then identify for each operation mode all the operational variables whose occurrence during that mode lead to accelerated failure. Such variables are referred to as stress factors. For instance, for a compressor one may define continuous operations and start/stop operations as two modes of operation whose duration or number of cycles is correlated to component failure. Under each of these modes, higher differential pressure may be regarded as a stress factor.

The CO2 HPWH is an extension of existing vapor compression technology to higher fluid density and pressure as well as transcritical operation. As a result, while a number of critical components for the CO2 HPWH are of relatively new design or operate under significantly different operating conditions, many of the components are borrowed from existing HVAC systems. In terms of reliability modeling, this means that for conventional HVAC components (such as the fan, evaporator, electrical system, and water pump), existing reliability data or reliability of similar components can be used. However, three major components: the compressor, electronic expansion valve (EXV), and defrost valve have been identified as having little or no reliability history.

To identify the reliability-related operation modes and stress factors, a FMECA (Failure Mode, Effects and Criticality Analysis) was performed. Based on this analysis, primary stress factors were identified. For the compressor, stress factors under continuous and start/stop operations were identified as the discharge to suction differential pressures and the oil temperatures. Likewise, for the EXV, the inlet to outlet differential pressures, the inlet temperatures, and the refrigerant mass flow rates were identified as stress factors. Finally, for the defrost valve, stress factors were identified as differential pressures and inlet temperatures.

A small number of dominant failure modes derived by expert analysis from the FMECA were selected for more in-depth analysis, with the remaining modes retained in a lumped, generic failure mode. Failure models were then identified for these components and modes. In our case they include wear and fatigue for compressor parts, fatigue initiation for EXV due to vibration, and sealing failure of the defrost valve due to material aging. Scaling models for these failure modes were established and key operational and environmental parameters were included in the models. Next, for the selected failure modes, we identified their relevant cycle definition (number of cold starts, hours of operation, number of start/stops, etc.). Using a time-to-failure distribution and a parametric scaling model (i.e. a model that calculates the equivalent number of cycles under different operating conditions) with unknown parameters, we can calculate the probability of system survival as a product of each failure mode's probability of survival. Since the survival probabilities will be functions of unknown scaling model parameters, a conservative estimate is made by maximizing with respect to unknown scaling model parameters the first year failure rate of the system, subject to constraints imposed by observing system and component survival under test conditions. These constraints are typically characterized by a lack of failures observed during qualification tests.

The solution provides an upper bound on FFR. Design of experiments can then be performed to find the smallest value of the upper bound (i.e. experiments that provide tightest FFR bounds). In other words, the solution is obtained by solving a Mini-Max optimization problem. The real advantage of the methodology is twofold. Unlike existing ALT methodologies, no detailed knowledge of scaling models is required. Better approximation of the shape of the scaling model yields more accurate results. However, under any circumstances, the methodology will be able to deliver an upper bound on the FFR. Secondly, it is not required to model all failure modes. Since a generic failure mode category is considered, finer decomposition only provides more accurate estimates.

We have derived mathematical proof that the solution to the Mini-Max problem for one failure mode, using linear-in-time scaling and no knowledge of the shape of the scaling model, is obtained by replicating through qualification testing the stress co-occurrences in the same cycle proportions as they occur during field operations. We have further proven that the bound on FFR provided by the methodology coincides under Weibull time-to-failure distribution with commonly known Weibull analysis formulas. This powerful result shows that, as flexible as the approach is in terms of not overly relying on the shape of scaling models and knowledge of failure modes, it is crucial to outline the stress factors completely so as to be able to replicate them through qualification tests. Starting with this boundary condition, we have extended the approach and have assessed its validity through extensive numerical studies.

Another advantage of the current methodology is that it allows optimal breakdown of an overall system reliability target to component level reliability requirements. This is advantageous relative to commonly used approaches in industry where overall system reliability is broken down into reliability targets for the components in a rather arbitrary manner, and test plans are developed to calculate confidence bounds on these individual reliability targets. The problem with this approach is that it is not clear at the outset what decomposition of reliability targets and confidence bounds on individual components would establish the required system reliability target at a desired confidence level and at minimum experimental cost. Since the problem is posed explicitly as an optimization in this approach, the most cost-effective breakdown and test plan is derived.

Example Result

As an example, we can decompose vapor compression system failure modes into three categories: compressor related, expansion valve related, and others. We further divide the compressor-related failures into three modes: seal/gasket failure, motor burnout, and valve failure. We make the assumption that these 5 failure modes (3 related to compressor, 1 expansion valve spindle fracture, and 1 related to the "other" category) are independent, i.e. the survival probability for the system over a period is the product of survival probabilities due to these failure modes. Although the failure category entitled "other" covers all other failure modes not included in the other 4 failure modes, it is expected to be dominated by control software failure. As we see later, information about the "other" failure mode is in many cases obtained largely by expert judgment.

Table 2.1 summarizes the failure modes and life models used. In this table t_{eq} are equivalent cycle counts over one year of service life, while t'_{eq} is any arbitrary equivalent cycle count for which the survival probability is to be calculated. As we see in Table 2.1, life models are assumed to be exponential distributions. These models are parameterized in such a way that θ_i is the negative log of survival probability due to failure mode *i* during first year of service life.

Component	Failure Mode	Cycle Count	Life Model
Compressor	Seal/gasket failure	Hours of operation	$P_S(\theta_1 \mid t'_{eq}) = \exp\left(-\theta_1 \times \frac{t'_{eq}}{t_{eq,1}}\right)$
	Motor burnout	Hours of operation at high temperatures	$P_S(\theta_2 \mid t'_{eq}) = \exp\left(-\theta_2 \times \frac{t'_{eq}}{t_{eq,2}}\right)$
	Suction valve failure	# starts at low temperatures	$P_S(\theta_3 \mid t'_{eq}) = \exp\left(-\theta_3 \times \frac{t'_{eq}}{t_{eq,3}}\right)$
Expansion Valve	Broken spindle	# close→open→close cycles	$P_S(\theta_4 \mid t'_{eq}) = \exp\left(-\theta_4 \times \frac{t'_{eq}}{t_{eq,4}}\right)$
Others	Predominantly control system related	Hours of operation	$P_S(\theta_0 \mid t'_{eq}) = \exp\left(-\theta_0 \times \frac{t'_{eq}}{t_{eq,0}}\right)$

Table 2.1: System decomposition, failure mode enumeration and life model selection

Table 2.2 shows how to summarize information for calculation of reliability bounds. An important data point needed is an estimate of the ratio T_{eq}/t_{eq}, i.e. the equivalent number of test cycles (for the given qualification tests) to the equivalent number of cycles during one year of service life. Generally there are a number of uncertainties in calculating these numbers, such as the average nature of the bin analysis used to derive the operating profile or uncertainty in quantification of the stress factors and exact parametric relations. For this reason, we later analyze the sensitivity of the results to wide range of variations in T_{eq}/t_{eq} relative to the baseline value used initially. These baseline values are presented in Table 2.2 together with number of planned component tests to extract information about each one of the failure modes.

Test	Failure Mode	Planned # of Tests	$\frac{\tau_{eq}}{t_{eq}}$
Compressor	Seal/gasket failure	4	0.5
	Motor burnout	4	2
	Suction valve failure	4	2
Expansion Valve	Broken spindle	4	4

Table 2.2: Information summary for calculation of reliability bounds

In addition to component tests, system experiments are planned to replicate the fraction of service life parameter φ. For example if we run the field trial units for 6 months under conditions similar to those experienced during service operations, then $\varphi = 0.5$. In the analysis that follows we fix φ to different values and obtain number of tests needed to obtain bounds on FFR at desired confidence levels. The system tests provide simultaneous information about all failure modes, including the “other” category.

Figure 2.1 demonstrates the results for 80% and 90 % confidence levels given a service life of 1 year ($\varphi = 1$) and choosing the T_{eq}/t_{eq} ratios presented in Table 2.2. This plot is obtained for a set of experiments that result in no failures. From this figure, we see that relative to the plot for 80% confidence level, the curve for 90% confidence shows an average upward shift of about 4% in *FFR*

Figure 2.2 and Figure 2.3 show sensitivity analyses of the calculated upper bounds to changes in the survival probability range of the “other” failure category as well as changes in the value of T_{eq}/t_{eq} ratio respectively. Such sensitivity analyses are important since information about the “other” failure category is most likely provided by expert judgment, and T_{eq}/t_{eq} ratios could be obtained either by expert interviews or using fixed-parameter acceleration models. From Figure 4 we see relatively small sensitivity to the “other” failure category survival probability range when the number of tests is small and larger sensitivity as the number of system tests increases. The opposite effect is observed in Figure 2.3, where the sensitivity is analyzed with respect to changes in T_{eq}/t_{eq} ratio. Only when the number of system tests is very small are the results sensitive to changes in T_{eq}/t_{eq} ratio.

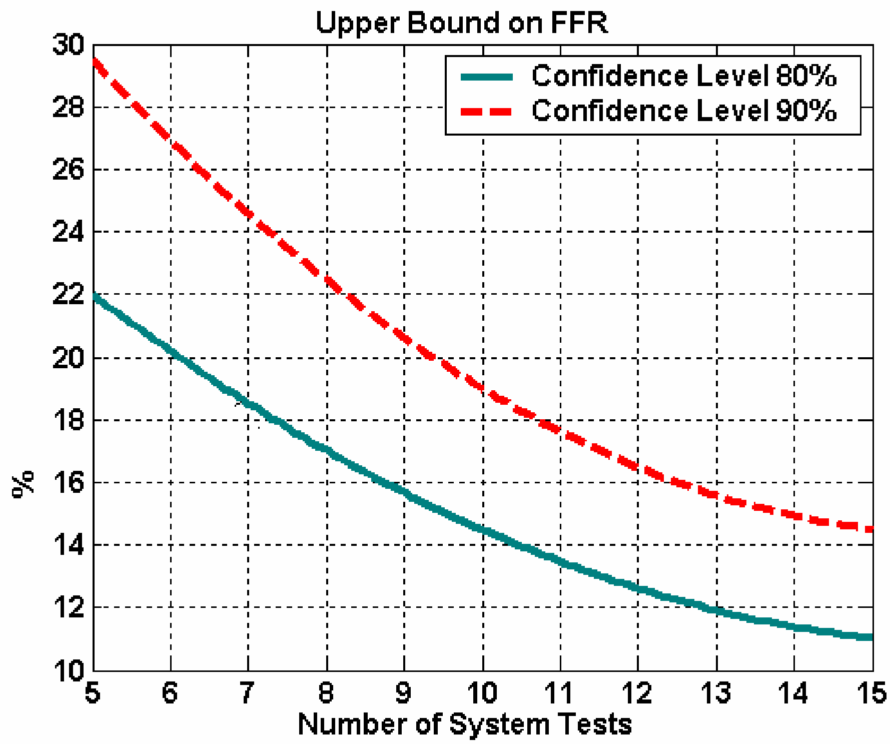


Figure 2.1: The two curves are obtained by setting T_{eq}/t_{eq} to the values shown in Table 2.2. Survival probability due to "other" failures is assumed to belong to the interval 0.88-0.98, and $\varphi = 1$ in both cases.

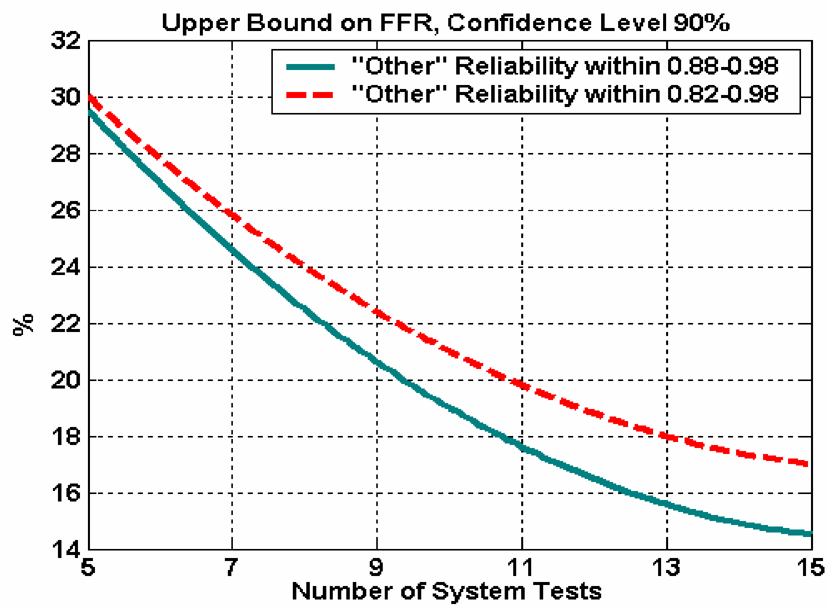


Figure 2.2: Both curves are obtained by setting T_{eq}/t_{eq} to values in Table 2.2 and $\varphi = 1$, but survival probability range due to "other" failures changes from 0.88-0.98 to 0.82-0.98.

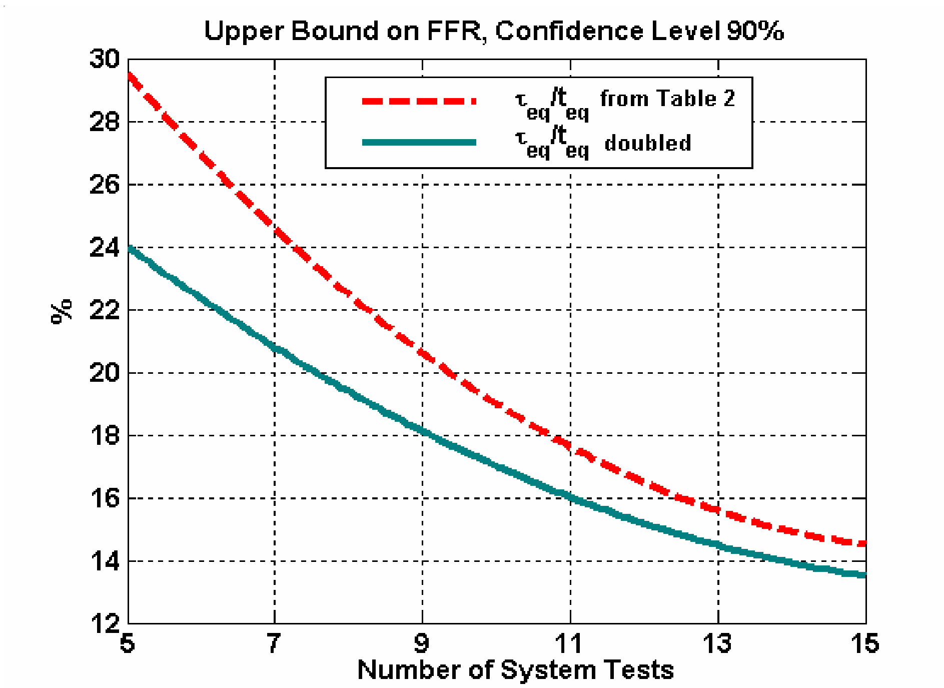


Figure 2.3: Keeping the survival probability range due to other failures to 0.88-0.98, the figure shows the effect of doubling T_{eq}/t_{eq} relative to the values in Table 2.2.

Figure 2.4 shows the upper bound on FFR calculated as a function of the number of field trial units. In obtaining this plot, we have incorporated modeling information about the operations of a CO2 HPWH as related to 4 failure modes (3 modes belonging to the compressor, one belonging to the expansion valve), the outcome of 5 sets of compressor and 2 sets of expansion valve qualification tests, and engineering judgment as to reliability range for other failures not modeled. Sensitivity analysis has been performed to obtain a range of upper bounds on FFR (the curve entitled "higher end of the ratios" corresponds to an average set of modeling assumptions, while the curve entitled "lower end of the ratios" corresponds to a conservative set of modeling assumptions). The results as presented in Figure 2.4 show a large sensitivity to model uncertainty only when the number of system tests is small. Only when total number of field trial units is small (around 5) does the FFR envelope obtained via sensitivity analysis have a relatively wide coverage. This coverage is around 4% in absolute value or 30% in relative terms for an approximately 100% change in modeling assumptions expressed as estimated equivalent number of cycles under qualification test conditions relative to actual field operations. This number is much less as the number of tested field trial units grows large (around 15). Hence when dealing with situations where very limited data is available, modeling accuracy may become an issue.

Further numerical studies were performed to assess the sensitivity of the new reliability estimation methodology to information quality and modeling assumptions. This case study considered a total of 6 failure modes, with 3 ascribed to the compressor, 1 to the defrost valve, and 2 major non-wear related modes associated with the integrated heat pump system. Thermo-dynamic stress factors (temperature and differential pressure) were assigned to 2 of these failure modes, with the remaining modes being excited by the total hours of operation and on/off cycles

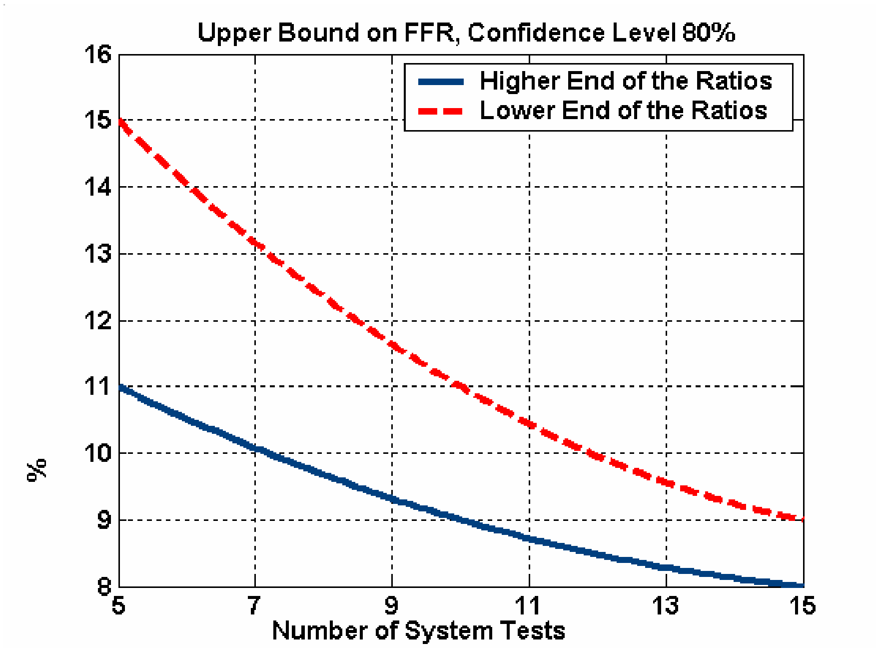


Figure 2.4: Upper bound on FFR as a function of the number of field trial test units

of the system. Weibull life models were employed to model fatigue initiation, wear, and aging during accelerated life testing. Data sources used in the case study include system thermodynamic models, empirical thermodynamic test data, averaged annual loading data, and expert opinion. Computations were performed to quantify the sensitivity of the worst case FFR estimates, generated using the new methodology, to variations in:

- scaling model structure and parameterization,
- number of units used for component or system tests,
- test conditions (intensity and duration),
- omitted information regarding important failure modes, and
- expert judgments used for scaling model parameter estimation.

This simulation demonstrated the strong sensitivity of estimated FFR to the number and duration of system-level tests, which indirectly also excite component-level failure modes, as compared to component-level tests, and also the importance of closely replicating the expected first year field operating conditions in any combination of system and component tests performed.

Task 3 – Understand Needs and Opportunities

The objectives of this task are fourfold:

1. Understand the relevant codes, standards and regulations across North America for heat pump systems in order to successfully meet these requirements for the CO2 system. These codes may be local, and a detailed understanding is required to enable broad commercialization of the system across North America.

2. Understand the current market in terms of products (including heat pump water heaters), segments, trends and forecasts, supply structure and applications. This allows for a better understanding of initial target segments for this technology.
3. Determine what installers and servicing customers are looking for in terms of performance, price, serviceability, and reliability from water heating products.
4. Determine market perceptions behind the lack of penetration of heat pump technology in order to understand the underlying barriers to entry of heat pumps in the North American market. These factors will be critical to determining the appropriate product for the North American market.

An independent market research firm, BSRIA Limited, was commissioned to perform data collection related to the above objectives. BSRIA determined to speak to market experts rather than convening mass interviews with the installers. Market experts chosen include:

- Signposting contacts; e.g. standards, organizations, and state companies
- Senior commercial contacts at major water heater manufacturers
- Senior commercial contacts at major heat pump water heater manufacturers
- Technical services contacts at the major manufacturers who are aware of installer and servicing requirements and also specific codes and regulations
- Specialized distributors targeted to understand the installer and servicing viewpoint
- Commercial installer organizations
- Industry bodies such as GAMA, research institutes, and consultants

BSRIA carried out this research in three phases. During each phase, BSRIA delivered interim and/or final reports to summarize their findings. Each phase is discussed separately below.

Phase 1 - Understanding of relevant codes, standards and regulation across North America

In Phase 1 the required NA codes for heat pump water heaters are presented in a single report "US water heating market update" which is outlined below. Government regulation of HPWHs is comprised of an overlapping network of federal laws plus state and local codes that are sponsored by different industry associations. This being said, UL certification is nationally recognized and is considered a pre-requisite for NA customers. Installation requirements also vary significantly, with the burden of code compliance typically on local installers that are licensed with their respective state or local authorities. Based on this finding, UTRC initiated work with the Conformity Assessment Service of UL to determine the requirements to achieve UL certification for this product as discussed in Task 5.

<u>US water heating market update- Part 1- the importance of relevant legislation, January 2004</u>

This report outlines the regulations and standards regime for the USA with particular regard to water heating appliances. A variety of regulations and US standards apply to different properties of the water heating appliance, which were categorized into eight areas by BSRIA as follows.

- *Electrical safety*

Federal law requires that electrical products intended for use in workplaces are certified by a "Nationally Recognized Testing Laboratory" (NRTL) and bear the mark of that organization. However, there are also laboratories and test houses outside the US that have agreements with the relevant US NRTLs (such as Underwriters laboratory) to be able to test products and affix the appropriate marks.

- *Electromagnetic compatibility*

Electrical products must also comply with electromagnetic compatibility (EMC) requirements enforced by the Federal Communications Commission (FCC). The FCC has accredited a large number of test laboratories both inside and outside the US to carry out the required tests. Accredited laboratories are listed on the FCC website.

- *Gas safety (if applicable)*

The standards for gas equipment are sponsored by the American Gas Association (http://www.aga.org).

- *Refrigerant safety (if applicable)*

Several regulations have been issued under section 608 of the Clean Air Act to govern the recycling of refrigerants in stationary systems and to end the practice of venting refrigerants to the air. There are also restrictions on the acquisition and use of certain refrigerants.

- *Hot water performance*

Instantaneous water heaters are generally rated in terms of the volume of water that can be supplied at a specified temperature given a specified supply temperature. Storage water heaters and indirect cylinders are rated on recovery time.

- *Pressure systems safety*

Pressure systems and components must generally conform to the 2001 ASME Boiler and Pressure Vessel Code. There are various exclusions for low pressure heating and hot water systems.

- *Hot water safety*

Workplaces are recommended by OSHA to hot store water at 60°C with 50°C available at the faucet to reduce the risk of Legionnaires disease. The risk of scalding at the point of use can then be reduced by installing thermostatic mixing valves.

- *Building and plumbing regulations and codes*

From 1972 the CABO (Council of American Building Officials) served as the umbrella organization for BOCA (Building Officials and Code Administrators International), ICBO (International Conference of Building Officials), and SBCCI (Southern Building Code Congress International), the major publishers of model building codes. In November 1997, it was agreed to incorporate CABO into the International Code Council (ICC).

ICC (http://www.iccsafe.org/) also includes a number of representative associations from the construction and property sectors such as:

- BOMA - Building Owners & Managers Association (www.boma.org)
- NAHB - National Association of Home Builders (www.nahb.org)
- NAA - National Apartment Association (www.naahq.org)
- NMHC - National Multi-housing Council (www.nmhc.org)
- AIA - The American Institute of Architects (www.aia.org)

It should be noted that in the USA, most standards are "sponsored" and ultimately published by sector based industry bodies. The American National Standards Institute (ANSI) acts as the national coordinating institution through which interested organizations may voluntarily cooperate in establishing, recognizing, and improving standards. ANSI does not test or approve products on its own. ANSI does certify a product when it is found, through examination by an independent laboratory, to meet the requirements of an approved standard.

Examples of organizations setting standards that are accepted by ANSI include ASHRAE, AFNOR, ASME, ASTM, IEEE and SA. Underwriters Laboratories (UL) produces nationally recognized standards.

Another influence on product selection is the Energy Star Scheme for new houses. It is a voluntary labeling program designed to identify and promote energy-efficient products, in order to reduce carbon dioxide emissions.

There is not a published Standard for 'Commercial Heat Pump Water Heaters'. However there does appear to be at least a draft Standard for 'Residential (domestic) Heat Pump Water Heaters'.

In summary, it is important that the National Standards are adhered to and UL Certification (or equivalent) is essential if the product is to be taken seriously. Many organizations specify UL Certification as one of their requirements.

Phase 2 - Understand the current market in terms of products (including heat pump water heaters), segments, trends and forecasts, supply structure and applications, and particularly why heat pump water heaters have not been successful in penetrating the market.

Phase 2 involves a series of interviews with commercial contacts at major manufacturers, technical services managers, service contractors, architects, key customers, and commercial installers. These interviews are used to generate a better understanding of the issues raised in aims (2) through (4) in the introduction. The reports delivered: "US market for water heating", "US market for residential and specialty air conditioning – water and ground source heat pumps", and "Commercial heat pump water heaters", are outlined below.

With the current US water heating market being around $2.5B, the total market of electric heat pumps in the US peaked around 1000-1500 units in the late 1990s and has decreased to 100-200 units per year. The variation is driven by availabilities of subsidies. BSRIA conducted interviews with utility companies as well as major water heater manufacturers, distributors and installers. Their primary conclusions are 1) hot water expenses are small so the high initial cost cannot be justified, and 2) the cost of lack of hot water is very high so reliability is a critical issue. These issues could potentially be overcome by marketing of a reliable unit by a name brand company willing to invest in a trained dealer structure.

US market for water heating, March 2004

Data from this report are summarized in Tables 3.1 and 3.2.

	Sold to COMMERCIAL			Sold to RESIDENTIAL			TOTAL			Volume share		Value share	
	Volume	MSP	Value	Volume	MSP	Value	Volume	MSP	Value	Comm	Res	Comm	Res
Electric													
Electric instantaneous	60,000	196	11.7	20,000	94	1.9	80,000	170	13.6	75.0%	25.0%	86.3%	13.8%
Storage residential	50,000	185	9.3	4,250,000	180	764.8	4,300,000	180	774.0	1.2%	98.8%	1.2%	98.8%
Storage commercial	49,300	1002	49.4	700	859	0.6	50,000	1,000	50.0	98.6%	1.4%	98.8%	1.2%
Total electric	**159,300**	**442**	**70.4**	**4,270,700**	**180**	**767.2**	**4,430,000**	**189**	**837.6**	**3.6%**	**96.4%**	**8.4%**	**91.6%**
Gas													
Instantaneous	15,000	650	9.8	115,000	593	68.3	130,000	600	78.0	11.5%	88.5%	12.5%	87.5%
Storage residential	65,000	235	15.3	5,435,000	230	1249.7	5,500,000	230	1265.0	1.2%	98.8%	1.2%	98.8%
Gas commercial	98,800	1,602	158.3	1,200	1,435	1.7	100,000	1,600	160.0	98.8%	1.2%	98.9%	1.1%
Total gas water heater	**178,800**	**1,025**	**183.3**	**5,551,200**	**238**	**1319.7**	**5,730,000**	**262**	**1503.0**	**3.1%**	**96.9%**	**12.2%**	**87.8%**
Oil													
Oil residential storage	530	550	0.3	43,470	499	21.7	44,000	500	22.0	1.2%	98.8%	1.3%	98.7%
Oil commercial storage	980	2,295	2.2	20	2,545	0.1	1,000	2,300	2.3	98.0%	2.0%	97.8%	2.2%
Total oil water heater	**1,510**	**1,683**	**2.5**	**43,490**	**500**	**21.8**	**45,000**	**540**	**24.3**	**3.4%**	**96.6%**	**10.5%**	**89.5%**
Cylinders and coils													
Indirect	14,600	550	8.0	97,400	493	48.0	112,000	500	56.0	13.0%	87.0%	14.3%	85.7%
Tankless coils	1,000	125	0.1	69,000	120	8.3	70,000	120	8.4	1.4%	98.6%	1.5%	98.5%
Commercial storage tanks	31,500	701	22.1	500	637	0.3	32,000	700	22.4	98.4%	1.6%	98.6%	1.4%
Total cylinders and coils	**47,100**	**642**	**30.2**	**166,900**	**339**	**56.6**	**214,000**	**406**	**86.8**	**22.0%**	**78.0%**	**34.8%**	**65.2%**
Hot water supply boilers (1)	**8,900**	**4,005**	**35.6**	**100**	**3,555**	**0.4**	**9,000**	**4,000**	**36.0**	**98.9%**	**1.1%**	**99.0%**	**1.0%**
Total	***395,610***	**814**	***322.1***	***10,032,390***	**216**	***2,165.6***	***10,428,000***	**239**	***2,487.7***	**3.8%**	**96.2%**	**12.9%**	**87.1%**

***Source:* BSRIA based on supplier estimates**

***Note 1.* Potable hot water only for commercial appliances**

Table 3.1: US water heater market analyzed by value and volume, 2003

	% share
New build	18%
Replacement	82%
Total	100%

Table 3.2: US water heating market analysed by type of installation

Commercial heat pump water heaters, February 2004

As BSRIA notes in this report, the current total market for heat pump water heaters in the US peaked at 1000-1500 units in 1997 and is recently estimated to be 200 to 300 units. More than 50% of these units are sold in Hawaii. Nearly all of the remainder are sold in the Southeast: Alabama, Georgia and parts of Florida. With the decline of the market in Hawaii, precipitated by the withdrawal of subsidies, the market may have shrunk to less than 100 units. There is no established distribution structure for these products.

The major players identified by BSRIA are: Applied Energy Recovery Systems (E-Tech), Colmac Coil, Heat Harvester, WaterWise A/C, Thermoplus Air, and Vacom Technologies.

The major factors affecting the uptake of commercial heat pump water heaters, as summarized by BSRIA, are high installed cost, reliability, availability of technical backup, complexity of the technology, and distribution / service channels.

US market for residential and specialty air conditioning, water and ground source heat pumps, December 2003

In this report BSRIA estimates the overall size of the water-source and ground-source heat pump market at 142,541 units for 2001 rising to 145,392 in 2002 as shown in Table 3.3. The definitions of heat pumps included in the study are:

- Water-loop heat pump: temperature-controlled water circulating in a common piping loop
- Ground-water heat pump: water pumped from a well, lake or stream
- Ground-loop heat pump: brine circulating through a subsurface piping loop

	Value US $ million	Volume units
ARI 320 (WLHP)		
Horizontal	51.1	55,859
Vertical		
<5t (17,060 BTU/h)	18.6	21,158
>5t (17,060 BTU/h)	20.4	4,746
Console	7.7	9,689
Other	7.8	7,414
Total	105.7	98,867
ARI 325/330 (GWHP/GLHP)		
Horizontal	12.5	10,128
Vertical	28.2	24,928
Other	4.3	4,200
Total	45.0	39,256
Speciality markets	16.3	7,270
Grand total	167.0	145,392

Source: BSRIA

Table 3.3: Water-source and ground-source heat pump market analysed by product type, 2002

Phase 3 – Determine what installer and servicing customers are looking for in terms of performance, price, serviceability and reliability from water heating products.

Phase 3 interviews were conducted with a group of 15 mechanical contractors and 10 consulting engineers widely distributed across the Continental US, resulting in the report "US commercial water heating" outlined below. Based on the interview results, it is shown that contractors and consulting engineers see the main considerations for their customers when choosing mechanical service systems and products as performance, reliability and price. Energy and maintenance costs are significant but of less importance.

Just under half of contractors and a majority of consulting engineers had some experience and understanding of non-air conditioning heat pump systems. Initial reactions to the product concept were generally favorable with the majority of respondents equally divided between those expecting to try the product when it becomes available and those expecting to use the product when it is established. The majority of respondents seemed unable to come to terms with the pay back proposition for their customers or had found great difficulty in selling this kind of proposition in the past. Contractors able to provide an answer suggested that customers would expect pay back on extra investment in energy savings made in six months or so.

It is clear that an entirely different approach in terms of product offering and support will be needed to establish a new heat pump water heating system as a viable mainstream product. Offering a "fit and forget" box, simple to install in place of or alongside a conventional water heater may be the best alternative. Contractors able to design, install, and maintain systems must be made easy to find, which means good education of the technology. The product itself must be very reliable and/or be supplied with a back up system.

U.S. Commercial water heating – contractor & specifier research, June 2004

Telephone interviews were carried out with 15 mechanical services contractors and 10 consulting engineers, of varying sizes, widely distributed across the Continental US. The scale of the research means that this should be regarded as a sample rather than a true representation of the range of different circumstances and attitudes likely to be encountered in bringing the product to market. It was agreed that research would be carried out amongst HVAC contractors and specifiers (consulting engineers), rather than the plumbing companies, currently installing most water heating systems.

As summarized in Table 3.4, contractors saw the main considerations for their customers, in choosing mechanical service systems and products, as performance, reliability and price, with energy and maintenance costs of lesser importance. Consulting engineers in Table 3.5 also strongly identified performance and reliability as the highest priorities, but a significant number of individuals thought that continuing costs of energy and maintenance were of highest importance.

Trade attitudes toward the choice of product/system also suggest that reliability is the overriding concern.

Contractors able to provide an answer to the importance of payback suggested that customers would expect payback on extra investment in energy savings made within approximately six months.

Combining cooling with the water heating function was found to add to the attraction of the product. Contractors indicated that the product would also be frequently installed as an "add-on" to an existing or traditional system.

Contractors' views of their customers' main considerations when choosing systems/products	Ranking	Mean Rating	% of respondents regarding as most important
PERFORMANCE – Can cope with hottest days / maximum occupation/use / meets maximum hot water demand	1	4.80	-
RELIABILITY – Comfortable temperatures always maintained / hot water always available	2	4.73	33.3%
INSTALLATION PRICE / FIRST COST	3	4.60	33.3%
ENERGY COST SAVINGS	4	4.27	13.3%
CONTINUING MAINTENANCE COSTS	5	3.93	6.7%
SERVICE BACK UP	6	3.93	-
GOOD FOR THE ENVIRONMENT	7	3.47	-
LATEST TECHNOLOGY	8	3.27	-
SPACE SAVING / LOCATION	9	3.20	-
OTHER	10	1.27	13.3%

Table 3.4: Contractor views of end user / customer attitudes to system/product choice

Consulting Engineers' views of their customers' main considerations when choosing systems/products	Ranking	Mean Rating	% of respondents regarding as most important
RELIABILITY – Comfortable temperatures always maintained / hot water always available	1	4.90	20.0%
PERFORMANCE – Can cope with hottest days / maximum occupation/use / meets maximum hot water demand	2	4.60	10.0%
ENERGY COST SAVINGS	3	4.10	30.0%
INSTALLATION PRICE / FIRST COST	4	4.00	10.0%
SERVICE BACK UP	5	3.90	-
CONTINUING MAINTENANCE COSTS	6	3.80	20.0%
SPACE SAVING / LOCATION	7 =	3.60	-
GOOD FOR THE ENVIRONMENT	7 =	3.60	-
LATEST TECHNOLOGY	9	3.30	-
OTHER	10	1.90	10.0%

Table 3.5: Specifier views of end user / customer attitudes to system/product choice

In summary we find that the current market for heat pump water heaters is very small and driven by subsidies, that the potential market is very large, and that market experts believe that first cost and reliability are the critical issues that need to be addressed for market acceptance. Operating costs are less important to these experts, who see the need for payback periods of well less than a year to justify additional first cost. This is primarily the result of the perception of low operating costs and high loss-of-service risk. From an expert point of view, a drop-in HPWH, preferably with a backup heater, is needed to succeed.

To date we have not evaluated the less tangible aspect of customer, rather than contractor, preference in efficiency and in using "green" products. With the current world discussion on the criticality of global warming and rising fuel costs, this factor could become significant in overcoming the current perceptions of the technology.

Task 4 – Create NA Prototype Specifications

The objective of this task is to create the component and system level specifications for the North American Phase I field trial units. These field trial units employ basic modifications to the existing European Union (EU) design heat pump prototypes to allow operation at 60Hz and to meet basic US codes and standards. Detailed specifications were developed for each of the following components.

- Accumulator
- Compressor
- Evaporator
- Pipe fittings
- Gas cooler
- Oil pressure switch
- Pressure relief valve
- Pressure transducer
- Refrigerant lines
- Service valves
- Defrost valve
- Electronic expansion valve
- Filter / dryer
- Water pump

Baseline product – UTC European CO2 HPWH prototype

United Technologies Research Center had been developing CO2 HPWH technology for several years before entering an agreement to minimize barriers to implementation of a North American product. Several CO2 HPWH prototypes have been produced and Carrier has sponsored a number of field trials across Europe. The North American product was based on these units, so the general specifications and performance requirements of the European HPWH are presented below and in Table 4.1.

- Outdoor ambient temperature range: -20 ºC to 46 ºC
- Water inlet temperature range: 1 ºC to 40 ºC
- Water delivery temperature range: 50 ºC to 80 ºC
- Dimensions: 207 x 108 x 133 cm (length x width x height)
- Unit weight: ~590 kg
- Electrical: 60 Hz / 460 V / 3 phase, 40 Amp max continuous draw
- Noise: 82 dB(A) maximum
- Refrigerant working pressures: max 83 bar low side, 138 bar high side
- Double wall counterflow heat exchanger suitable for sanitary applications
- System designed to meet pressure safety factor of x3 with appropriate UL required safety features

Air Temp	Inlet Water Temp	Outlet Water Temp	Unit Power Draw	Heating Capacity	Heating Capacity	GPD Equiv	Steady-State COP
[F]	[F]	[F]	[kW]	[kW]	BTU/hr	gal/day	
-4	50	140	11.5	23.8	81,276	2,597	2.07
14	50	140	12.8	32.4	110,645	3,535	2.54
32	50	140	13.6	42.6	145,477	4,648	3.14
50	50	140	14.5	50.6	172,797	5,521	3.49
86	50	140	15.4	68.9	235,291	7,517	4.46
104	50	140	15.2	76.8	262,269	8,379	5.06

Table 4.1: Baseline design performance specification for EU CO2 HPWH prototypes

The major proposed changes from the EU units are as follows:

1. Compressor displacement: 50.8 cm^3 (from 62 cm^3)
2. Fan: 60 Hz motor
3. Water pump variator: 460 V / 60 Hz input & 400 V / 50 Hz output
4. 24 VAC supply transformer: 460 V input & 24 V output
5. Low pressure relief valve: Setpoint of 82.7 bar (lowered from 93 bar)
6. Filter/dryer: Burst pressure rating of 413.5 bar minimum
7. Water temperature limit control: Less than 85 ºC

The first item is the most significant design change from an overall system perspective. By decreasing the displacement of the compressor by ~22% (from 62 cm^3 to 50.8 cm^3), the rate of displacement remains virtually unchanged as we move from a 50 Hz to a 60 Hz power source. This implies that the overall system mass flow and capacities will remain unchanged, having minimal impact on the design of the other components. The 50.8 cm^3 displacement compressor is currently a model available from the manufacturer, and is therefore expected to have virtually no impact on commercialization feasibility. Items 2-4 are specifically related to the differences in voltage/frequency available in the US (460 V, 60 Hz, 3f) versus the EU (400 V, 50 Hz, 3f). Items 5-7 are specifications that have been developed to meet UL requirements. More specifically, items 5 and 6 have been included to meet the x3 factor of safety required for fatigue testing as outlined in UL 1995. Item 7 is a general water delivery temperature requirement for potable or sanitary water systems.

Based on Task 3 findings that UL listing is a prerequisite to market acceptance, the units were designed to be compatible with UL certification requirements although they will not have a UL mark. The process of obtaining UL certification has been discussed with representatives of UL. Initial assessments indicated that it would not be possible to meet UL 1995's standard for x5 factor of safety for static pressure tests unless a significant re-design of major components was undertaken. However, the current system design has the capability to meet the x3 factor of safety for the fatigue test as outlined under section 61A of the standard. In addition to pressure safety, there are various other UL requirements that cover everything from compressors and electrical motors to electrical wiring, material compatibility, and unit closures. Details of specific component eligibility for UL listing is discussed in more detail in Task 5.

Task 5 – Component Development

Components for the North American HPWH system are either extracted directly from the European prototype system or modified to meet US power supply and regulatory requirements as discussed in Task 4. Component development during this project fell under three broad categories: component modification from EU specification, demonstration of component reliability, and meeting requirements for UL listing.

Component modification

The modifications required for prototype testing were relatively straightforward. The only functional changes in the NA system were forced by the change in supply voltage and frequency. The original European design is intended to meet European CE and PED requirements, so the modifications were designed to maintain all of the original component characteristics given the new power supply.

To ensure system flow (and thus pressure) and control characteristics remained identical; the compressor displacement was reduced by shortening the stroke by the 60/50Hz ratio, a change initially prototyped by UTRC but eventually adopted as a product by the compressor manufacturer. This required changes to the crankshaft and connecting rods, both highly stressed components important to reliability. The speed of the compressor is increased by the same ratio, which has widespread effect in the compressor. This change increases the inertial load on the reciprocating parts and tends to increase oil temperature through increased shear, but it also improves the rod bearing oil film. The motor, which is at least partially oil cooled, may also experience reduced reliability because of this increase in speed. All of these reliability challenges must be validated through qualification testing. Compressor development and reliability for the North American HPWH was strongly intertwined with, and leveraged by, development work on the 50Hz European compressor. As development was driven by reliability, these issues are discussed in Task 6.

Other modifications were less complicated. The water pump was maintained at the same speed as the 50Hz unit by substituting a variable speed drive compatible with 60Hz. The control input vs. pump flow relationship is thus maintained and no control modification is necessary. The evaporator fan motor was replaced by a similar US-based motor designed and UL listed for 60Hz. Because the fan speed is higher at 60Hz a US-based fan, from the US Aquasnap product that was the basis of the CO2 HPWH chassis, was adopted to maintain similar airflow characteristics in the phase 2 units. The other modifications to EU specifications were addressed by substitution of components or subcomponents that have relatively little effect on the operation or control of the system, such as the pressure relief valve, filter/dryer, and control voltage transformer. The only control modification directed exclusively by the NA specification was a modification to the tank thermistor setpoint. None of these modifications were expected to have a significant effect on system reliability.

UL listing

Through the work in Task 3, it was found that UL listing is a de-facto requirement for successful market introduction of commercial heat pump water heaters, and the majority of effort in Task 5 was directed toward this goal.

UL listing of the CO2 HPWH can be achieved by meeting three basic requirements: all components of the system must be UL listed, the complete system must undergo operational tests for UL compliance, and a submittal document must be provided to UL for final approval. UL listing is a goal that can only be reasonably accomplished when a device reaches the manufacturing stage. For this project our intent was to systematically remove any barriers to achieving UL listing so that listing could be granted very soon after a decision to manufacture was reached.

The first requirement for obtaining a UL listing for the HPWH is to ensure that each component used in the system is UL listed. To be practical this is a requirement that must be adopted by the component manufacturer. While it is feasible for a packager like Carrier to have a sourced component listed, any change implemented in the part by the manufacturer will initiate relisting and the yearly compliance testing becomes the responsibility of the lister rather than the manufacturer. Because the CO2 HPWHs are currently being developed for a European market most of the components are not UL listed. On the component level, our approach was as follows:

- Identify all non-UL listed components
- Request UL listing for non-listed components from the existing component suppliers, or
- Inquire about alternate UL listed components from that vendor that meet specifications, or
- Obtain a comparable UL listed component from a different vendor, or finally
- Perform or outsource required UL testing.

Often a vendor will list components for a product already in the US market. If our specified component is not listed, then we have asked the vendors to have the component listed. Because most component manufacturers would require some minimum of sales to take on the burden of listing, and Carrier has not yet committed to producing the heat pump water heater, few vendors are willing to apply for a new listing. In some cases an alternate component that is already listed will meet the specifications. The next alternative is choosing a new supplier based on availability of listed components. This is not a step taken lightly as manufacturer-supplier relationships are difficult to build. The best case is one in which the Carrier factory can specify another active supplier with a listed component alternative. Finally, UTC can choose to take responsibility for listing the component. Assembled subsystems built by Carrier may be listed by Carrier rather than the subsystem component manufacturers. The external cabinet is an example of this. Also, Carrier or UTRC could apply to list a component before the manufacturer has a commitment for sufficient production volume to justify listing themselves. This would be a temporary measure to demonstrate that barriers to listing could be overcome with the intent that the component manufacturer would take over the listing once production volumes were adequate. An example would be burst testing of an unlisted component to validate if listing of this component would ultimately be possible.

The current status of UL listing is partially complete. The bill of materials for the heat pump has been reviewed for existing UL listing. At this time only a small percentage of the components are listed. UL files exist for the following components:

- SWEP double wall gas cooler
- ECO evaporator coil
- Parker filter / dryer
- Texas Instruments high and low pressure switches
- Kriwan compressor motor protective module and sump heater

Extensive discussions were held with the bill of material manufacturers to initiate UL listing of their respective components. In general the manufacturers were unwilling to list components because Carrier had not made a firm commitment to production of the HPWH. Significant discussions were held particularly with the compressor manufacturer Dorin, the EXV manufacturer Saginomyia, and the water pump manufacturer Salmson, but in no case was a submittal initiated.

In a few cases an alternative supplier was pursued. Grundfoss supplies a UL listed water pump that meets specifications, and Parker has UL listed pressure relief valves. Discussions with Carrier in Europe to consider these alternate suppliers were also not ultimately productive.

Having tried these courses of action, we determined to perform critical testing at UTRC to ensure that there were no substantial barriers to UL listing once manufacturing commitments could be made. To be certain this action would be effective, representatives of UL were hired as consultants to guide us through the details of the process. The requirements for UL listing of heating and cooling equipment are spelled out in UL Standards 984 and 1995. The primary requirements can be summarized as:

- Demonstration of safety margin of pressurized components
- Demonstration of margin to electrical overheating
- Water intrusion testing of electrical components
- Dielectric strength testing
- Fluid compatibility testing
- Corrosion testing

Our highest priority was to determine if the major components would be capable of meeting the UL pressure requirements which are quite different from the European PED requirements. Next we addressed the issues of appropriate UL design for the system electrical box and the potential for water intrusion. Motor insulation dielectric testing was also performed for the current components, but more comprehensive testing will be performed as a standard part of compressor qualification. Finally, standard Carrier fluid compatibility and corrosion testing for the European product was deemed adequate to assure satisfaction of UL standards.

Initial pressure testing of the gas cooler, evaporator coil, accumulator, and the service module (which includes the expansion valve, service valves, filter/dryer, defrost valve and all associated lines) was performed. In addition, testing of the complete compressor shell was performed to ensure that at least low side pressure requirements could be met. The results are summarized in Table 5.1.

Component	UL Code	Requirement	Status	Comments
Compressor (low side)	61, 61A	3571 psig	OK	compressor shell tested to 3600 psig
Compressor (high side)	61, 61A	5971 psig	Uncertain	
Ref line - comp to GC	33.1, 61, 61A	5971 psig	OK	x3.8, calculated
Ref line - comp to defrost	33.1, 61, 61A	5971 psig	OK	x4.4, calculated
SWEP GC	61, 61A	5971 psig	OK	> 3, tested
Ref line - GC to filter	33.1, 61, 61A	5971 psig	OK	x4.4, calculated
Filter	61, 61A	5971 psig	OK	> 3, tested
Ref line - filter to EXV	33.1, 61, 61A	5971 psig	OK	x4.4, calculated
EXV	61, 61A	5971 psig	OK	> 3, from Saginomyia
HP service valve (Parker)	61, 61A	5971 psig	OK	> 3, tested
HP Isolation valve (Parker)	61, 61A	5971 psig	OK	> 3, tested
LP service valve (Parker)	61, 61A	3571 psig	OK	> 3, tested
Ref line – defrost to EXV	33.1, 61, 61A	3571 psig	OK	x5.3, calculated
Ref line - EXV to evaporator	33.1, 61, 61A	3571 psig	OK	x3.7, calculated
Evaporator (dist. to header)	61, 61A	3571 psig	OK	x5.4, from ECO
Ref line - Evap to accumulator	33.1, 61, 61A	3571 psig	OK	x3.7, calculated
Accumulator	33.4, 61, 61A	3571 psig	OK	x3.8, tested
Ref line – accum. To comp	33.1, 61, 61A	3571 psig	OK	x3.7, calculated

Table 5.1: Component pressure requirements and margins

All of these components with the exception of the compressor high pressure side were shown to withstand at least 3x overpressure through calculation, testing at the manufacturer, or testing at UTRC. UL 1995 requires that listed components withstand at least 5x overpressure, which only the evaporator coil ultimately met. UL code does allow listed components be tested to only 3x overpressure if they can pass fatigue testing which specifies 250,000 cycles of operation at 3x pressure without failure. During negotiations with UL an exemption to this fatigue test requirement at 3x overpressure was discussed and eventually granted. With this exemption in place the testing summarized in Table 5.1 appeared to support eventual UL listing from a pressure test standpoint. However, this exemption was later rescinded on further review within UL. Further progress on UL listing of these components will depend on successful completion of fatigue testing at high and low side pressures or respecification of the components.

To ensure UL compatibility of the electrical box the manufacturer Siemens was consulted. The original box was designed by Siemens in Europe and UL standards were not part of the specification. A review by Siemens US highlighted a number of details that were not UL compliant. Siemens drafted a UL compliant electrical box design and began manufacturing a prototype albeit with a very long lead time. Based on later project priorities no UL compliant electrical boxes of the Siemens design were ultimately procured, but it was shown that this component design can meet UL standards at a producible cost.

In addition to these component tests, the system was also successfully rain tested, a UL requirement for all outdoor units. The test involved a rain simulation on the unit, one hour with the fan off and one hour with the fan on, with the rain directed at the most vulnerable part of the unit in terms of water penetration into electrical equipment. At the end of test, all the electrical equipment was visually inspected for water penetration. In addition, a dielectric test was conducted for all motors.

A submittal for complete system UL listing has been substantially completed, although successful submittal would still depend on the completed listing of each component and inclusion of the UL listing references in the system listing. Although far from complete, we expect that the system as designed will meet UL standards with little further modification.

Task 6 – Component Reliability Assessment

Component reliability is typically addressed through component qualification testing. Carrier maintains proprietary protocols for qualification testing of every component needed to fabricate a heat pump. The uniqueness of the CO2 HWPH comes from the very high pressures and from the fluid itself; its interactions with the components, joints and lubricants. The European HPWH is derived from the fully qualified Carrier Aquasnap product. Qualification protocols for components not exposed to these unique challenges can be applied directly to the CO2 HPWH components with no modifications. For instance, the cabinet, fan and motor, water pump and drive, water piping and water sensors can be treated as fully qualified based on previous testing.

Other components have been developed significantly in the EU CO2 HPWH prototypes and are considered low risk for the North American program. Pipes, brazing processes and fittings have been developed extensively and this experience was applied directly to the NA prototypes, particularly the phase 2 units. The heat exchangers, pressure switch, transducers, and thermistors are rated by the manufacturers for these pressures and chemical exposure and are considered relatively low risk even though they have not been specifically qualified for these conditions using Carrier protocols.

Three major components have been identified as being significantly different, in technology, design, or operation conditions, relative to Carrier's standard product offering: the compressor, the electronic expansion valve (EXV), and the defrost valve. Of these components, only the compressor design is different for the NA prototypes. Component development for reliability was therefore primarily supported by the EU HPWH activity and the NA project could leverage these results. The resources of the NA project component reliability efforts were thus focused primarily on evaluating the risk of lowered compressor reliability directly related to the design changes made for 60Hz operation. In addition, some root cause analysis activities on EXV failures were included in the NA project, particularly those on later failures of the last generation EXV that were only experienced in the NA prototypes.

Component Test Plans

Before testing component reliability, the proprietary Carrier qualification test plans for the compressor, EXV and defrost valves were reviewed using the reliability modeling methodology developed in Task 2. Information from the current qualification tests was extracted for scaling model development and reliability assessment. Modifications were then proposed to the current qualification and test plans to add new test/measurement items and implement alternative ways of conducting the tests so that the scaling models for failure modes and model parameters could be established from these tests.

For the compressor, less than 1% FFR (or 99% reliability) for the first year is required to meet the total system 7.5% FFR target. As illustrated by Table 6.1, this requires 69 compressor units to be tested, with no failure, to achieve even 50% confidence. The current qualification test plan contains items for normal tests, accelerated tests, mechanical tests and environmental tests with a total of 58 units. The number of samples for normal and abusive tests is 33. Parameters to be measured include suction pressure, discharge pressure, suction temperature, discharge temperature and oil temperature. More test units are clearly needed to satisfy the currently proposed component-level reliability targets of 1% FFR for compressor. The following additional measurement items and an alternate test approach were proposed.

1. Start-stop testing for bearing touchdown damage needs to be augmented by adding one more full lifecycle unit test to address bearing wear during cyclic operation
2. The material wear loss of each part should be measured at 500 hour intervals for normal continuous tests and every 100k cycles for normal start/stop tests, and at the end of any accelerated test.
3. Finished compressors should be reassembled to run through the rest of the test items under normal and abusive conditions. A spectral analysis may be needed to define loading levels and numbers of cycles for each test items to be representative of the field loading conditions and provide more reasonable, rather than over-conservative, prediction. Depending on number of units that can be tested through the spectrum, different reliability and confidence levels can be verified as dictated by Table 6.1.

Confidence \ Reliability	50%	90%	95%	97%
50%	1	7	14	23
80%	3	16	32	53
90%	4	22	45	76
95%	5	29	59	99

Table 6.1: Number of samples needed to achieve specific reliability/confidence levels

For the EXV, the Carrier standard qualification test requires that the initial release probability of survival is to be a minimum of 50% with 90% confidence, improving to 97% with a confidence of 80% within 15 years. To demonstrate this, a defined number of complete cycles are required. For the minimum reliability requirement, only 4 samples are required to be tested with no failure. For the desired long-term reliability 53 units must be tested without failure as shown in Table 6.1.

The proprietary test plan as written will fully satisfy the minimum reliability requirement. The reliability goal was set with respect to the most severe loading condition (100 bars) while in the test plan, only 8 units will be tested under this loading condition. These 8 units test will give 81.8% reliability with 80% confidence or 75% reliability with 90% confidence. Additional no-failure tests under lower loading conditions will provide better data for updating the reliability model. More ideal options are:

1. Test all units at 100 bars. This will ensure 95% reliability with 80% confidence or 92.6% reliability with 90% confidence under the most severe loading condition; or
2. Test all units under a load spectrum that will be representative of the field loading conditions, as an example, A cycles at no load, B cycles at 10bars and C cycles at 100bars. No failure during these tests will provide the same reliability and confidence levels and are less over-conservative.
3. After completing unit tests for corrosion, humidity, and potentially even hydrostatic pressure, put these tested units through endurance testing as described in options 1 or 2 above. Testing these additional units will validate the reliability and confidence levels within a fraction of a percent of the desired target of 97% reliability with 80% confidence.

Additional measurement items may include:

- Taking fluid temperature measurements to improve the scaling model
- Continuing corrosion testing after prescribed number of hours of salt spray until failure occurs (i.e. surface corrosion observed)
- Measure vibratory acceleration at different differential pressures and fluid temperatures to establish scaling model parameters for fatigue failure caused by vibration

Negotiations with the EXV supplier were initiated to determine the most practical combination of tests.

For the defrost valve, the Carrier standard requires that the probability of survival after a certain number of cycles occurring over 75,000 hours of operation shall be at least 95%. Assuming 80% confidence, this requires that 31 units to be tested without failure. The Carrier standard specifies endurance tests, corrosion tests, hydrostatic tests, and humidity tests. The same considerations apply as for the EXV except for vibration measurement. Additional measurements to improve scaling models identified as important by the FMECA should include valve leakage and modulus of the sealing material at the end of each test.

Component reliability testing

As discussed in Task 5, the major changes in the system hardware were made to the compressor. Increasing the compressor speed from 50Hz to 60Hz will increase stress on the compressor subcomponents in two primary ways. First, the 20% higher speed will cause higher acceleration and inertia effects in the NA compressors, so inertia forces and imbalance effects will be greater. Cylinder pressures will not be substantially affected, so pressure forces will remain similar. However, the shorter stroke of the NA compressors will result in lower forces in the direction normal to piston motion, that is, lower piston side loads and lower piston skirt wear per mechanical cycle. Second, the higher speed will result in 20% more mechanical cycles per unit time than the European compressor, so wear on parts will be generally greater per unit time in the North American compressor. The higher speed in the NA compressor will result in greater oil film thickness in the journal bearings, but also will result in higher viscous power loss in the bearings. The valves will operate more cycles per unit time as well in the NA compressor. This will affect wear and impact noise. Because flows and pressures remain similar, related loading on the valves will not change.

The proprietary qualification testing for the CO2 HPWH compressor for the European application is planned to be a joint effort between the vendor Dorin and UTRC. The test plan for the European product was developed before the DOE NA agreement was put in place. However, qualification testing for this product has not yet been executed. Field trial results in Europe in 2002 and 2003 showed that the compressor connecting rod big end bearings were not reliable. UTRC and Dorin executed design changes that resolved the field trial issues, but it became clear from this work that high temperature conditions in the southeastern US would be very challenging to the compressor bearing durability.

Virtually all of the compressor testing performed for the European prototypes could be directly leveraged for the NA prototype, so relatively little actual compressor development was performed as part of this contract. To ensure that any eventual North American CO2 HPWH product did not force further compressor qualification testing, the Carrier compressor qualification plan was revised to account for the possibility of 60Hz operation in addition to the 50Hz in the original plan. This change, although straightforward, increases the difficulty of passing the test protocol. Some initial (pre-qualifying) compressor reliability testing was performed using the NA design 60 Hz compressor to ensure that this design could be expected to pass the Carrier protocol. This testing was fully in parallel to European development.

The primary issues that surfaced that can be directly related to the compressor modifications were:

- A continued lack of connecting rod bearing success at the “maximum load” condition, a high ambient air and water temperature condition where oil viscosity is minimized while pressure loading is still very high. Failure is defined as bearing-journal material transfer.
- Insulation failures in the primarily oil and air-cooled motor windings
- Suction valve fatigue failures

Careful testing of both compressor versions demonstrated that connecting rod bearing problems were not significantly more likely in the NA unit. The failures were determined to be a result of inadequate viscosity, and thus lubrication film thickness, for the existing load. We believe that the increases in inertial forces and film thickness effectively cancel out in the NA compressor resulting in statistically similar bearing lifetime. Development of solutions to this problem, primarily bearing / lubricant composition studies, was performed under the European compressor workscope.

Motor insulation failures on the other hand were limited almost entirely to 60Hz compressors. Previous refrigerant/oil compatibility testing was successful, so overheating at the 480V 60Hz condition appeared to be the root cause of these issues. For the development of the NA phase 2 prototypes a refrigerant-to-oil heat exchanger was specified. This device brings the oil sump temperature down significantly with beneficial effects on the primarily oil/air cooled motor and also on the bearing oil film thickness. This change has also been incorporated into the design of the European HPWH, although no European field trial units were so equipped.

Finally, occasional suction valve failures were noted on both NA and European compressors. These failures were found to result from high cycle fatigue. Although the NA compressor accumulates more cycles than the European unit and this is a contributor to HCF failures, root cause analysis demonstrated that the issue was one of process quality of the valve manufacturing process. Valves that met specification did not fail in either compressor.

So the primary revision to the system derived from 60Hz compressor testing was the incorporation of an oil-to-refrigerant heat exchanger to ensure both sufficient motor cooling and bearing film oil viscosity at the highest anticipated ambient temperatures.

EXV qualification for the European prototype was performed by the supplier Saginomyia according to a proprietary Carrier test plan. This valve required no adaptation for use in the NA prototype, so the qualified valve was applied to the phase 1 prototypes with no further testing. As noted in Task 2, an assessment of the EXV qualification test plan using the reliability model identified a number of changes that could be implemented to improve the component expected reliability. These changes were adopted into the test plan for future valve versions but were not included in the initial qualification tests performed by Saginomyia for the European product.

Once the phase 1 North American field trials were in service a number of EXV failures began to occur both on European and NA field trial units. Rigorous root cause analysis demonstrated that flow-induced vibration (FIV) at particular valve openings were resulting in internal fatigue failures. These failures took some time to show up because operational time at the very small valve opening of concern accumulated very slowly, but once they began virtually all units failed and required refit. This failure mode is particularly interesting because the FMECA performed for the Task 2 reliability model identified FIV as a significant risk. The recommended changes to EXV qualification testing resulting from the reliability model review addresses operating spectrum testing and instrumentation with accelerometers that might have identified this problem before the valves were used in the field.

Concurrent testing by UTRC and Saginomyia, funded by the European FT effort, led to modifications of the valve design. A new generation of EXV was utilized in the phase 2 NA field trial units. This change was implemented as a field fix without full vendor qualification. Perhaps not unexpectedly, some of these valves have also failed in the NA field trial units. Inspection of these failures, which so far are restricted to North American units and not retrofitted European field trials, shows fatigue failure of a spring fitted to raise the natural vibration frequency of the valve internals. This issue has not been resolved at the present time.

Defrost valve qualification testing was also performed by the vendor. Again, an assessment of the defrost valve qualification test plan using the reliability model identified a number of changes that could be implemented to improve the component expected reliability. These changes were adopted into the test plan for future valve versions but were not included in the initial qualification tests performed by Danfoss for the European product. In the case of the defrost valve, no field issues other than control algorithm flaws have surfaced so no further development of the European specification valve for the NA prototypes has been undertaken.

Data from the testing performed, both 60Hz and 50Hz systems, were reviewed for incorporation into the reliability models. Reliability issues encountered particularly on the compressor and EXV forced design changes during the entire course of this contract. Because of the limited number of tests performed on any one component design, collected data were of very little benefit to the reliability model.

Task 7 – NA/EU Prototype Testing

Three prototype, or phase 1, CO_2 heat pump water heaters were fabricated and installed in the field. Figure 7.1 shows the schematic of the instrumented units, the physical component layout, and the appearance of the units. The functional specifications of these units are effectively identical to the European HPWH units and they were manufactured in the same Carrier factory in Montluel, France. Upon arrival in the US we first performed some component and control upgrades that were developed after the components were sourced from the factory. These were primarily upgrades to the compressor developed during component testing for both EU and NA products. Also, CO_2-to-oil heat exchangers, part of the phase 2 specifications, were backfit to the phase 1 units. These upgrades are noteworthy primarily because it means that all of the phase 1 units were subject to compressor removal and disassembly / reassembly at UTRC, a process that could affect reliability in either direction.

Once modified, we performed a series of tests to determine any variation in operational characteristics relative to the European unit. In particular, one unit was installed in a psychrometric facility at UTRC and tested to determine the optimum refrigerant (CO_2) charge and to verify control characteristics, particularly the optimum pressure mapping that controls the EXV setting, and thus the compressor discharge pressure, needed to achieve optimum efficiency from the transcritical cycle at the given temperature conditions. Once settings were verified, all three units were installed in the test facility to undergo a complete functionality test, which verified the operation of safety devices (both hardware and software), unit operation at the extreme ends of the outdoor ambient air temperature envelope, high temperature soak startup,

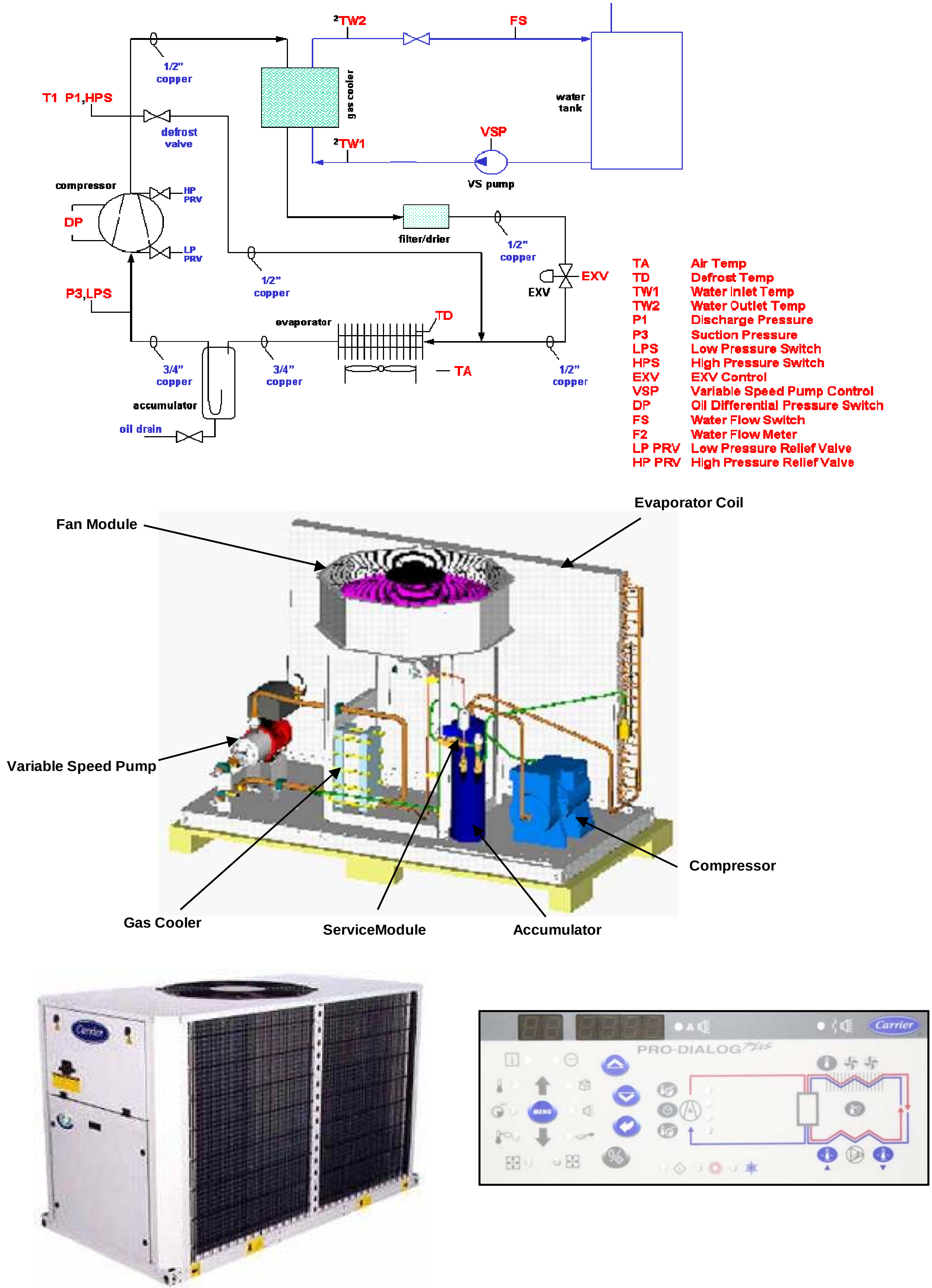


Figure 7.1: Schematic, layout, external appearance, and control panel of the HPWH prototypes

and defrost. Measurements were also taken to benchmark the performance of the unit. Finally, the remote data monitoring, archival, and retrieval system were tested. The commissioning test plan is outlined below:

- Calibration of sensors for COP measurement
- System static pressure check to 1.2 times maximum operating pressure
- Low and high pressure safety switch test
- Defrost test (at 0 °C, 100% RH)
- Hot soak test (at 46 °C)
- -20 °C to 46 °C ambient temperature operation
- CCN (Carrier Comfort Network) interface test
- Remote data collection functionality

Representative psychrometric room test data for the phase 1 field trial units are presented in Table 7.1 and Figure 7.2. The steady-state performance was approximately 20% below that of the prototype European units tested before the NA work began. Although it was not clear at the time this issue was generic to compressors destined for both EU and NA prototypes produced during a certain time interval. The problem was later identified as leakage caused by poor quality control on valve plate flatness, a critical performance specification because of the high valve differential pressure. Testing with valve plates verified to be in specification resulted in a 20-30% improvement in compressor isentropic efficiency and unit COP relative to earlier units.

OAT	Low P	High P	SST	EWT	LWT	Water ΔT	Water Flow	Capacity	Electric Power	COP
[F]	[psig]	[psig]	[F]	[F]	[F]	[F]	[gpm]	[BTU/hr]	[kW]	
5.6	291.9	1246.1	0.4	49.3	141.6	92.3	1.58	73,175	12.45	1.72
31.6	401.6	1380.5	19.4	61.3	139.5	78.2	3.14	122,983	14.48	2.49
51.9	476.4	1458.2	30.2	64.2	142.2	78.1	4.00	156,791	15.60	2.94
77.1	589.7	1509.7	44.6	69.1	138.3	69.2	5.85	202,738	16.79	3.54
106.3	642.3	1472.2	50.5	63.9	137.0	73.1	6.37	233,243	16.85	4.05

Table 7.1: Steady-state performance of unit 12Z318131 at different ambient temperatures

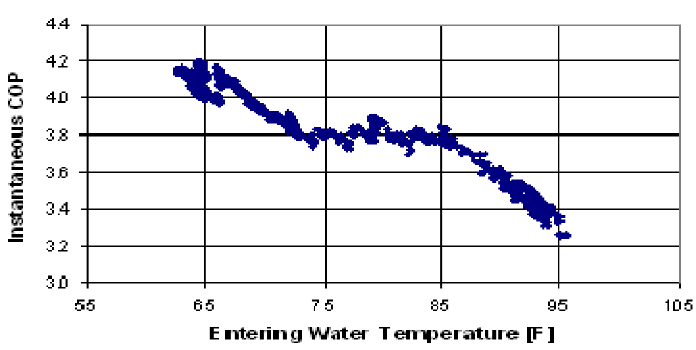


Figure 7.2: Transient response of COP to entering water temperature, unit 12Z318130

Three field trial sites were identified for phase 1, two sites in Alabama and the third at the campus of UTRC in Connecticut. The two units in Alabama were placed indoors while the unit at the UTRC campus was placed outdoors. The three applications are described in detail below.

The first field test site, shown in Figure 7.3, was located in an aerospace manufacturing facility in Alabama, where the unit was used to provide hot water for an industrial process called paraplast. The paraplast process requires periodic hot water supply at temperatures greater than 80 ºC to support a manufacturing process that produces thermoplastic aerospace components. The primary purpose of this field trial was to subject the heat pump unit to large numbers of on/off cycles in order to collect reliability data under extreme conditions. The hot water demand of the field trial facility averages to 3800 liters per day with peaks of 27 liters per minute. The facility is also equipped with an electric water heater with a capacity of 125 kW. The field trial heat pump was installed to provide base load capacity and preheating. This integration reduces the power demand on the electric heater and at the same time guarantees the hot water supply even if the CO2 HPWH fails.

As an additional benefit, the CO2 heat pump supplied cold air from the evaporator to the building where it was installed. The building has to be air-conditioned constantly to remove heat from the paraplast process. The cooling capacity comes as a “free” benefit and reduces the load on the chiller system of the facility. The average daily energy consumption of the original system with electric heater alone is approximately 280 kWh. The projected energy consumption of the modified system with CO2 heat pump pre-heat and electric heater was 154 kWh per day, 54 kWh provided by the CO2 heat pump and 98 kWh supplied by the electric heater. The cooling capacity provided to the building averages to 50 kWh per day if a 10 SEER chiller system is assumed as baseline. The annual energy savings of the CO2 heat pump thus adds up to approximately 38,300 kWh. The energy savings translate to a reduction of electricity costs of $1,150 per year at an average price of electricity of 3 cents per kWh. While the cost benefits for this application are limited, the operational experience gained during the field trial provided valuable input to the reliability data of the unit. This unit began operation in December 2004 and was decommissioned in December of 2006. In this time the unit operated 754 hours with 1670 compressor starts.

The second field trial site, shown in Figure 7.4, was located outside the cafeteria of the United Technologies Research Center in central Connecticut. The cafeteria serves approximately 400 meals per day and has an annual water usage of approximately 5400 m3. The CO2 heat pump was integrated with the original steam heater for the hot water supply. Before the heat pump installation, cafeteria hot water was generated by the building steam supply. After the installation of the heat pump, the building steam supply provides peak load and backup during maintenance caused outages of the heat pump. The heat pump is controlled by a thermistor in a 454 liter storage tank. If the hot water level in the storage tank falls below the set point, the HPWH unit will start generating hot water. A substantial value for this site was the proximity to the laboratory, as any failures can be quickly diagnosed and resolved for this unit. The unit installed at UTRC campus was subject to extreme outdoor conditions. The reliability of the electrical components and the water loop was tested with respect to freezing and other inclement weather conditions. This unit began operation in November 2004 and was decommissioned in January 2007. In this time the unit operated 2810 hours and experienced 3324 compressor starts.

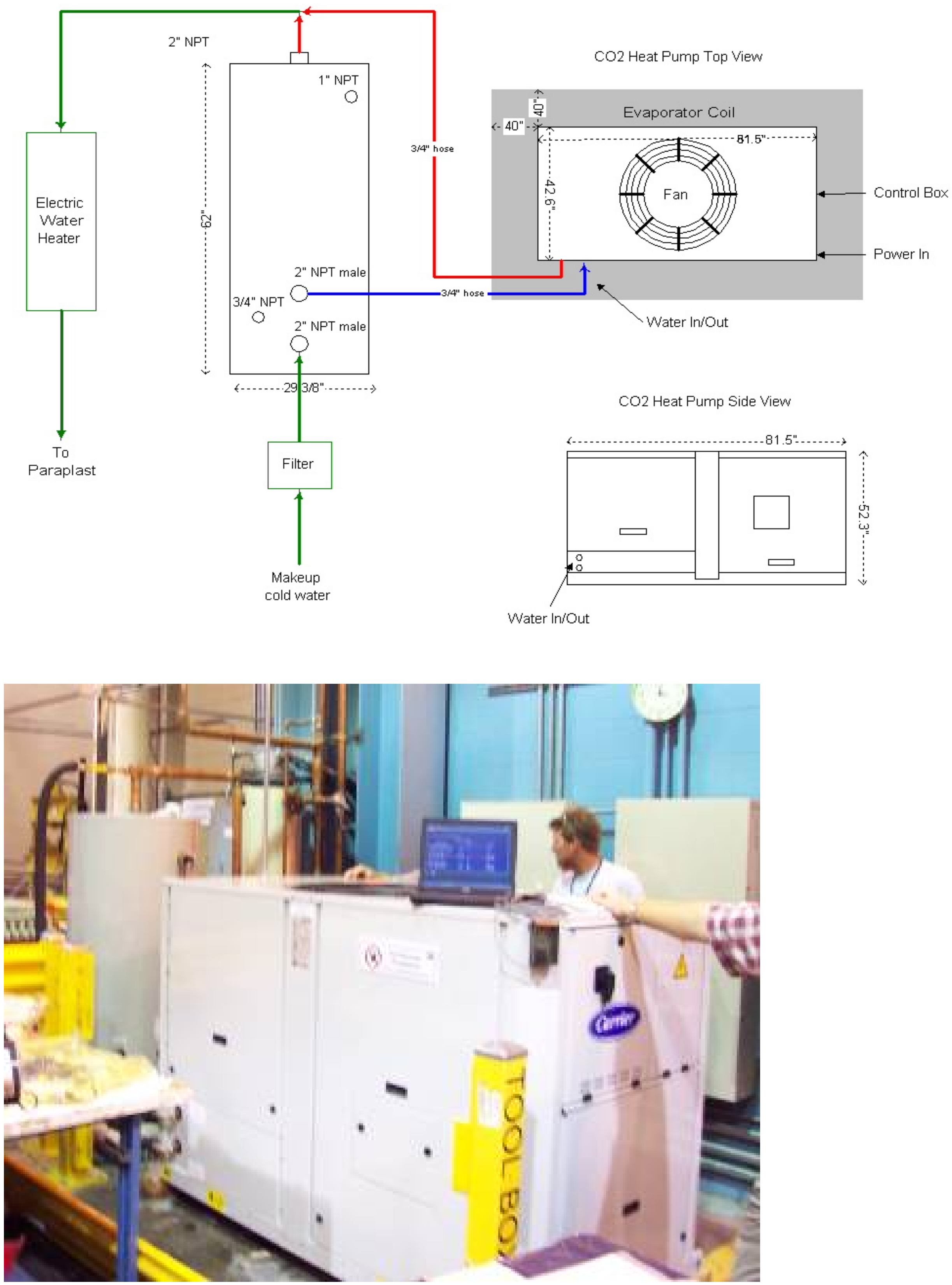


Figure 7.3: GKN Aerospace installation schematic and picture

Figure 7.4: UTRC cafeteria outside installation

The third field trial site, shown in Figure 7.5, was at the Druid County Hospital laundry facility also located in Alabama. Laundry facilities in hospitals have a considerable demand for sanitary hot water. The field trial site operates in a 16-hour shift and utilizes 250,000 liters of hot water per day. The water is heated by steam, which is generated in a dual fuel furnace with approximately 80% efficiency, and stored in an open tank. A float in the tank opens the water line from the city supply if the tank level falls below the set point. Fresh water is heated in a once-thorough steam heat exchanger. A pump circulates the water in the tank through this heat exchanger if the tank temperature falls below set point.

The CO2 heat pump was integrated into the system to provide a fraction of the base load of the facility. A solenoid valve controls the water supply from a separate heat pump storage tank. The valve opens if the water level in the open tank is below the set point and if the water temperature in the storage tank is above a certain level. The float in the open tank first opens the solenoid from the heat pump storage tank before it opens the water inlet to the steam water heater. This set-up guarantees that the system takes full advantage of the cost and energy savings from the CO2 heat pump, while providing reliable hot water supply to the laundry facility. The primary purpose of this field trial was to collect performance and reliability data of the unit under constant load conditions. It was also chosen to stress the unit on constant high ambient temperature operation caused by the proximity of the large steam boiler to the HPWH unit. Because of the utility price structure in the region of the hospital, the cost for gas, oil, and electricity are approximately 4 cents per kWh independent of the energy source. This price structure is one of the reasons why the southeast of the US is one of the primary traditional markets for electric heat pumps. The annual energy cost savings average to $10,400 at an average cost of 4 cents per kWh gas/oil/electric. The heat pump was installed in April 2005 and was decommissioned in December 2006. In this time the unit operated for 3985 hours and experienced 3091compressor starts.

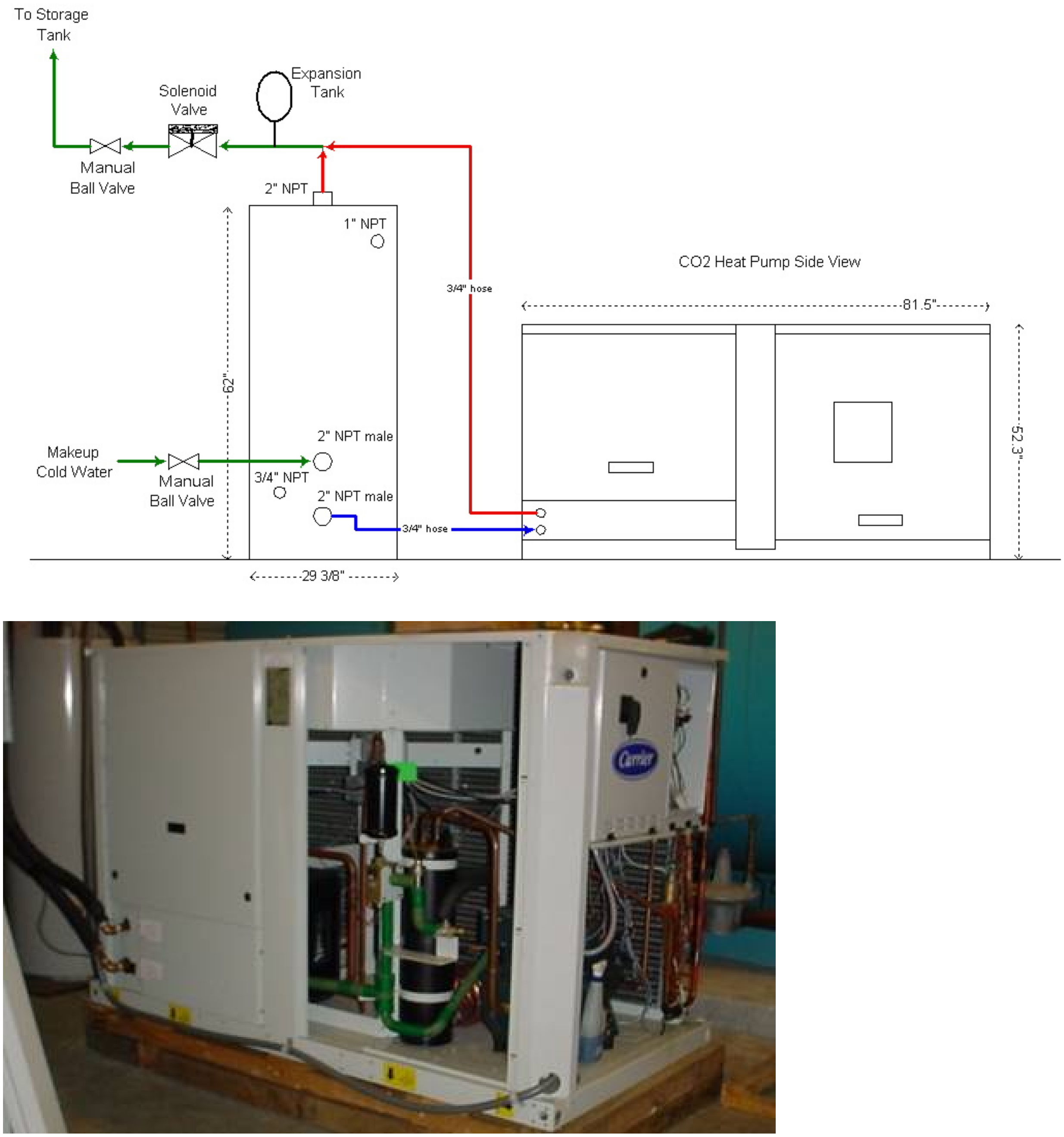


Figure 7.5: Installation at Druid County Hospital

Table 7.2 summarizes the installation and operational data for the phase 1 field trial units. These units accumulated 7549 operational hours over 70 unit-months. A variety of contractors installed the units based on the recommendations of the field trial participants. UTRC used in-house facilities staff for the installation, resulting in the lowest cost, while the other sites employed outside contractors. Carrier Commercial Services participated in the GKN installation as an observer and in the DCH installation as general contractor.

Site	Compressor Run Hours	Compressor Starts	Contractor	Cost of installation
12Z318131 (UTRC) UTRC Cafeteria, CT	2810	3324	UTRC	$5,185
12Z318130 (GKN) GKN Aerospace, AL	754	1670	Quality Metal Fabricators	$7,077
12Z318132 (DCH) Druid County Hospital, AL	3985	3081	NC Morgan (CCS)	$8,605

Table 7.2: Operational and installation data for the phase 1 field trials

Performance data were collected using remote data acquisition systems based on the Carrier CCN monitoring platform. Each unit sent data over a phone or Ethernet connection to a remote database server and was added to a relational database developed for the project in collaboration with database consultant MindTree. As discussed further in Task 9, the remote connections and database server were not reliable and the actual data retrieved are limited. Although much data were lost, some data were retrieved for each of the field trial units.

Figure 7.3 shows an example of this data, in this case for the GKN Aerospace site early in the unit life. Although only basic condition and performance data are shown, a total of 45 measurements and control states were acquired approximately twice a minute.

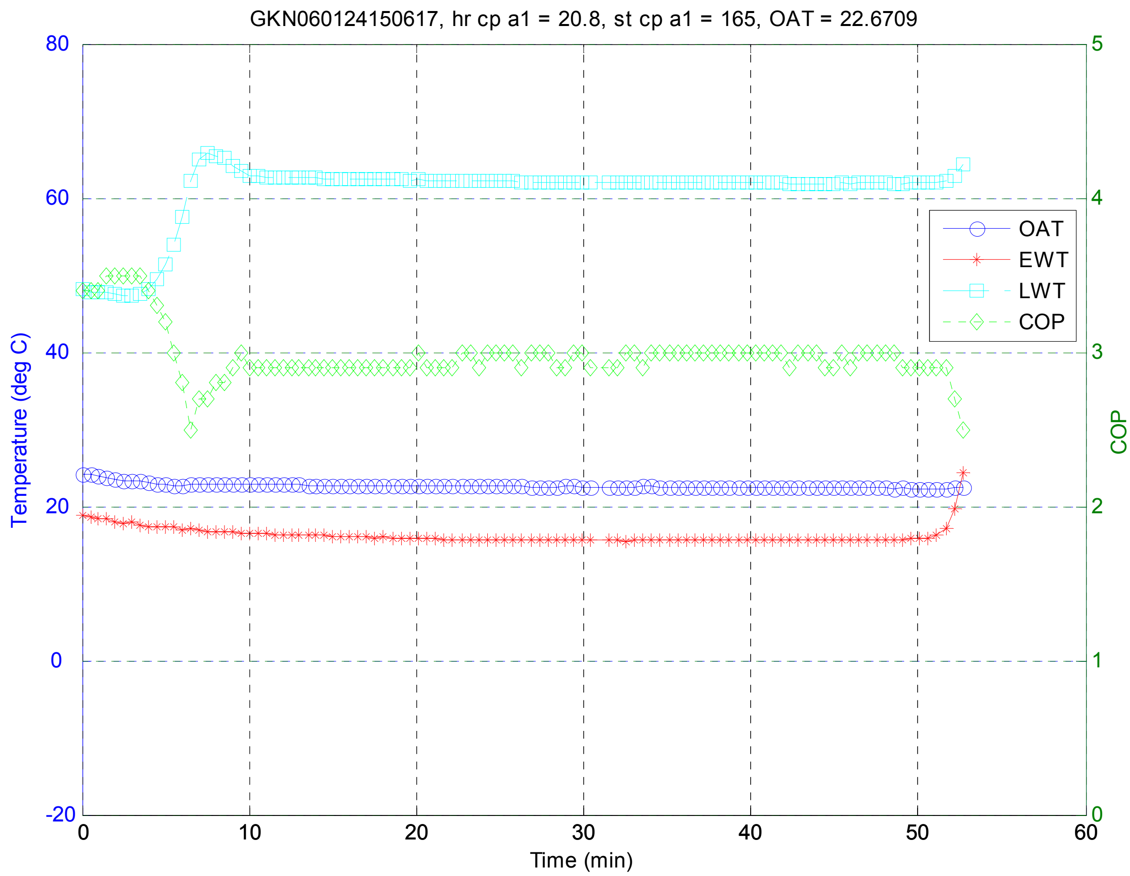


Figure 7.6: Example of field trial condition and performance data

Averaged performance data are compiled in Table 7.3 for the three phase 1 units and compared to the design performance originally presented in Table 4.1 in Task 4. It is clear that the observed performance is less than the design performance, although in most cases comparable to early psychrometric room testing (shown in Table 7.1 earlier in Task 7) except for the case of the DCH trial. As discussed in Task 9, it was later learned that the water flow meter on the DCH unit was calibrated to read only 50% of the actual flow. Correcting for this brings the performance in line with the other units. As discussed earlier, these data represent poor compliance with the valve plate specification. Points substantially more than 20% below design are believed to be primarily the result of low system charge caused by gradual and difficult to detect leakage that was commonplace on the phase 1 units, although the UTRC unit appears to be below average in any case.

Site	Ambient air temperature OAT (ºC)	Entering water temperature EWT (ºC)	Leaving water temperature LWT (ºC)	COP	Design COP (deviation)
UTRC Cafeteria	3	20	69	2.0	2.6 (-30%)
(outside)	8	22	70	2.1	2.8 (-33%)
	14	13	66	2.5	3.4 (-36%)
	22	21	68	2.4	3.4 (-42%)
	35	28	69	2.4	3.7 (-54%)
GKN Aerospace	22	18	62	2.9	3.6 (-24%)
(inside)	25	25	61	3.0	3.7 (-23%)
	25	33	62	2.7	3.3 (-22%)
Druid County Hospital	22	12	62	1.5 (3.0)	3.8 (-27%)
(inside boiler room)	31	12	62	1.7 (3.4)	4.3 (-26%)
	34	13	63	1.8 (3.6)	4.4 (-22%)

Table 7.3: Relative performance of the phase 1 field trial units

Task 8 – Controls Upgrades and Shakedown Testing of NA/EU Prototype Units

The objective of this task was to modify the existing EU CO_2 HPWH control logic as required to ensure proper operation of the field units. In line with this objective, an experimental evaluation of the suitability of the existing EU controller for NA systems was first undertaken. 418 data files from 5 European FT units were reviewed for controller operation. From this, three priorities for control system updates were identified.

- Improve defrost logic
- Update the optimal pressure map for the NA specification
- Address poor control response and sensitivity to operation condition

The defrost logic needed modification to prevent high-pressure shutdowns on entry into defrost. Through a series of experiments, the optimal temperature for termination of the defrost cycle was identified to be 195F. This setting prevents high pressure trips.

The optimal pressure map needed to be updated to identify the true optimal pressures for the NA hardware configuration rather than those for the EU unit. The efficiency of the CO2 HPWH is a strong function of the accuracy of the optimal pressure setpoint over the operating envelope. The hardware differences driving this need include a shortened compressor stroke, addition of an oil cooler, fan model changes, and the 20% increase in rotational speeds due to the 50Hz to 60Hz power conversion. A large change in the map developed for the EU units was not expected.

Using data collected in a series of performance experiments that spanned the entire operating envelope of the system, a COP versus pressure relationship was extracted, and a new optimal performance equation was derived. The new pressure map differs significantly from that inherited from the first generation EU units, particularly at high air temperature conditions where a 300-450 psi reduction in optimal pressure was observed. The root cause of the map variances has not been identified but it likely rooted in the same high pressure suction valve leakage that was determined to reduce the performance of all of the NA and many EU HPWH field trial units.

Architectural changes to the continuous (PID) controller structure and tuning were needed to make the system more responsive to setpoint changes and less sensitive to the operating point (predominantly driven by the EXV position). To address this issue, a model-based controller design effort was undertaken. This involved the design and execution of a comprehensive set of system identification experiments, the development of a nonlinear transient model from the collected data, the successful cross-validation of this model against normal operating data, and the design of a gain-scheduled PID control algorithm for the system.

The dominant control-related difficulty affecting the EU field trial units was the interaction between the temperature and pressure controls. Slow response to pressure variations had been observed, resulting in large pressure variances about the set point. This highlighted a need for faster EXV actuation. Large variations occur in leaving water temperature and compressor discharge pressure under otherwise steady environmental conditions. This indicates a strong need to replace the existing decentralized PID architecture with a multivariate one. The first step was to perform detailed system identification testing to develop a global nonlinear model of the CO2 HPWH system.

System identification testing of the modified 60kW unit at UTRC was carried out via the introduction of pseudo-random binary signals with switching frequencies < 0.5 Hz to the control inputs for outside air temperature (OAT), entering water temperature (EWT), expansion valve (EXV) setpoint and variable speed pump (VSP) setpoint. The resulting transient operating data was used to estimate linear time-invariant transfer functions to predict compressor discharge pressure (P1), compressor discharge temperature (T1), leaving water temperature (LWT), and coefficient of performance (COP).

Sensitivity of the system to OAT and EWT was found to occur primarily below 0.05Hz, which is partially attributable to the large transport delays affecting these signals. At higher frequencies, 0.05Hz-1Hz, the EXV and VSP dominate at operating points close to the boundary of the operating envelope. Generally, VSP sensitivity changes marginally with operating point, while EXV sensitivity varies by several orders of magnitude over the operating envelope. This suggests that PID gain scheduling will be required if EWT and OAT disturbances are to be effectively suppressed.

The low-speed (1Hz) data collected from this testing was used for estimation of discrete time linear transfer functions, which approximated the system dynamics over an envelope of operating conditions. Accuracies of +/- 2% were obtained for all test conditions except those involving low (-20°C) ambient temperatures, where an accuracy of +/- 2.6% was ultimately obtained. A response surface fit was then used to combine this set of linear models into a global nonlinear model of the system. The latter was used to design a gain-scheduled PID control algorithm for the NA field trial units.

Initial testing of the gain-scheduled control algorithm demonstrated instability, most likely because the available steady-state map of valve flow coefficients vs. EXV position was not sufficiently accurate to support dynamic compensation for nonlinearities in the system response to the EXV. At this time resources were not available to complete testing of the new gain-scheduled control algorithm before the control algorithm had to be finalized to allow release of the phase 2 prototypes. The final control algorithm was thus a PID-based algorithm derived from the EU prototypes, with a series of detail updates to improve performance and reliability developed on the NA phase 1 units. The following upgrades were incorporated into the original EU control design and ultimately implemented in the phase 2 units:

- Update Sequential Control Settings
  - Startup, Warm-Up, Steady, Shutdown, Fault Detection and Diagnostics
- Add Control Algorithm Enhancements:
  - Add Static Compressor Discharge Temperature Estimator
  - Add Static Non-linear Pre-Filter for EXV Gain Scheduling
  - Implement Defrost Mode Cascade Temperature & Pressure Control
- Re-tune PID Loops for Robust Performance:
  - EXV (Warm-Up, Steady Mode & Defrost Mode Pressure Control)
  - Variable speed pump (Water Control)
- Re-Estimate Optimal Pressure Map

Task 9 – Field Trial Reliability Assessment

The field trial reliability assessment has three components: identifying operational failures, performing root cause analysis of failures, and updating the reliability growth model to evaluate our current estimate of FFR.

Each field trial unit was equipped with Carrier Comfort Control Network (CCN) capability and a communication board. The proprietary CCN system allows monitoring of all control sensor outputs as well as all calculated parameters, alerts, and alarms. In most cases CCN is accessed on-site by a service technician. However, the NA HPWH field trial units were designed to allow remote access to CCN through modem or Ethernet connections so operational data and alarms could be continuously retrieved and stored to a database for analysis. When this system was working, we could track performance of units daily and were immediately aware if a unit shut down. In many cases the alarm could be reset remotely and the unit would return to service. If not, we could survey the data stream, see what had caused the alarm, and the team sent to repair the unit could take exactly the parts or tools necessary to bring the unit back into service.

These remote access units tested very well, but increasing security issues at UTRC and at host sites resulted in the placement of firewalls and other security measures that continually interfered with data retrieval over the term of the field trials. Because of this, data retrieval was intermittent at best from most units with the exception of the UTRC unit that was connected directly into our Ethernet. Regardless of this, some operational data was retrieved from each unit, although the last unit placed at the Hilton Garden Inn was only accessible by connecting on-site. The primary impact of lost communications was delay in servicing of units in alarm. Host site personnel routinely monitored the systems and informed us of alarms, and also helped diagnose issues remotely, but this was not as efficient as remote access.

In addition, the database and analysis software did not prove to be very functional. The vendor contracted to build the database, Mindspring, completed the software, loaded data into the database, and locally demonstrated the capability of the software to generate reports of unit operation. However, we were not able to reliably access the database or retrieve these reports from UTRC. All data presented in this report therefore were reduced individually.

Operational failures

The operational histories of the field trial units are shown in Figure 9.1. Green indicates that the unit was operational, red; the unit was shut down, and pink; a unit that was either shut down by the host facility (no failure) or was not reporting data and suspected to be down. It is clear that a number of failures occurred on the units, and that no unit escaped failure. The longest operational stretches were on the order of 3-4 months. Also of note is the length of down time, particularly of the units outside of Connecticut. In 2006 the resources to repair the units were limited. Maintenance was typically delayed until more than one unit was down to make a service trip more cost effective. All failures were repairable in one or two days at most, so accumulated run time could have been much higher if timely maintenance had been possible.

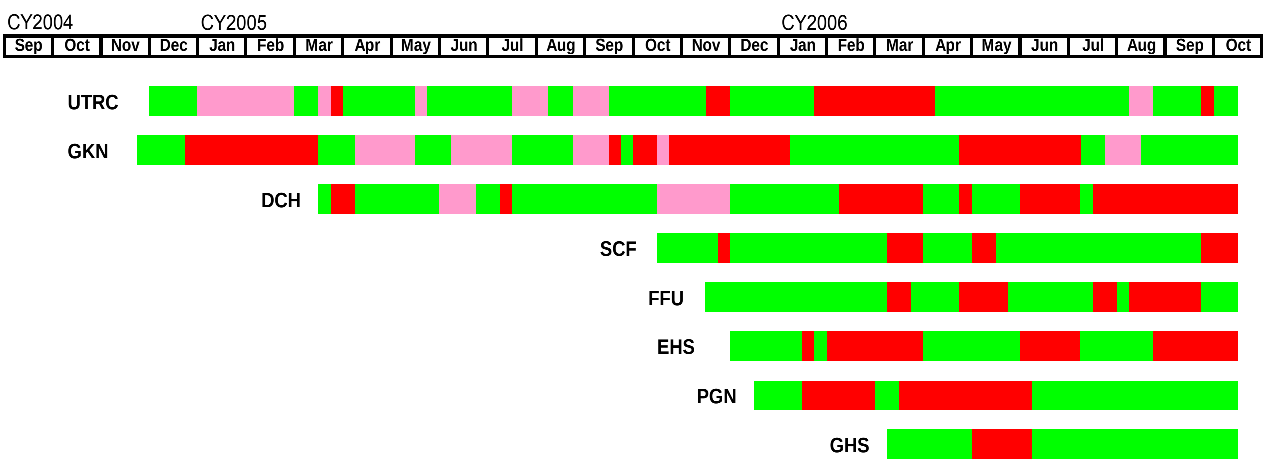


Figure 9.1: Timeline representation of field trial unit availability

Commissioning issues were observed on most units and these were tracked separately from in-service failures not related to the installation. On all of the units the as-received low-voltage signal connections at the sensors/transducers/switches and the control board were not robust. Intermittent loss of connectivity, witnessed during unit checkout tests and during field trial operation, caused numerous alarms and alerts especially in the phase 1 units. Instances of incorrect system wiring were also noted: a reversed fan, switched pump drive wires and an improper transformer. Although not pertinent to the product, we also noted that two water flow sensors had incorrect calibration resistors and read only 50% flow. Several leaks were observed which could have occurred in manufacturing or shipping, and several leaks that developed shortly after the units were put in service might have also resulted from inadequacies in the as-received condition of the units. Two service modules supplied with the phase 2 units were inoperable on initial test with no clear reason. Finally the control algorithm was initially not well tuned, allowing large setpoint overshoot and occasional trips during commissioning and early stages of operation. Specific commissioning failures are listed in Table 9.1.

Unit	Failure
12Z318131 / UTRC Cafeteria, CT	Loose wiring in permissive loop
	Control box wiring loose
12Z318130 / GKN Aerospace, AL	Control box wiring loose
12Z318132 / DCH Birmingham, AL	Leak in compressor external suction piping
12Q503329 / Superior Fish, MS	Control box wiring loose
	Fan wiring reversed
	Pump drive not operational
	Water fitting not tightened
12Q503332 / Fairfield University, CT	None noted
12Q503327 / EHS, Birmingham AL	Service valve packing loose
12Q503328 / Shari's Restaurant, OR	EXV not operational
	Service module leak
12Q503331 / Hilton, Glastonbury, CT	Supplied with incorrect control transformer
	EXV not operational

Table 9.1: Commissioning failures experienced in the field trial units

Operational failures of the field trial units are summarized in Table 9.2. The 45 overall failures can be divided into several categories: major components, sensors and wiring, leaks, and control algorithm issues. Each category will be discussed separately

Major component failures observed involved the compressor, the EXV and the pump variable speed drive. Root cause analysis with the manufacturer was completed for the first two failures. The single compressor failure occurred after this particular compressor was disassembled at UTRC to incorporate some last minute design improvements developed from the EU HPWH program. It was traced to a failure to fully torque the connecting rod bolts. Once the compressor was rebuilt we found no oil return to the compressor sump. This was traced to debris blocking the accumulator oil return. Because there was no sign of lack of lubrication in the compressor it was determined that the accumulator was blocked by debris from the initial compressor failure.

Unit	Failure	Comments
12Z318131 / UTRC Cafeteria, CT	24V breaker trip	Control algorithm
	Discharge pressure transducer	
	Compressor continuous run	Control algorithm
	Leak at burst disc threads	Manufacturer issue
	Compressor failure	Improperly installed rod bolt (UTRC)
	No compressor oil return	Accumulator blocked with debris
	Compressor motor temp trip	Loose wire in protective loop
	PRV leak	
	Suction pressure transducer	Connector damaged from service work
	PRV opened	
	EXV failure	Gen 1 EXV faulty
	PRV leak	
12Z318130 / GKN Aerospace, AL	Leak in service module	Probable flaw as-received
	Water tank thermistor	
	Defrost valve leak	Probable flaw as-received
	Filter/dryer leak	Probable flaw as-received
	EXV failure	Blockage from defrost valve repair
	Fan not running	Control algorithm
	Leak at compressor fitting	
	Water thermistor bushing corroded	Incorrect bushing material installed on-site
	High pressure trips	Setpoint on LWT too high
12Z318132 / DCH Birmingham, AL	Evaporator coil leak	
	Pressure switch damaged	Overtorqued on installation
	Unit running without fan (2 failures)	Control algorithm
	Tank thermistor failed	Scaling
	Leak at discharge thermistor	
	EWT thermistor failed	
	PRV leak	
	EXV failure	Gen 1 EXV faulty
12Q503329 / Superior Fish, MS	Pump drive failure	
	EXV failure	Gen 2 EXV - new issue
	EWT thermistor failure	
	EXV failure (apparent)	Not verified or repaired
	Flowmeter gasket failure	Water loop freezing pushed gasket out
12Q503332 / Fairfield University, CT	Pump drive failure	
	Repeated high pressure alarms	Initially replaced EXV, but unit still tripped. Loose wires on main contactor
12Q503327 / EHS, Birmingham AL	Leak at braze on evaporator coil	Probable flaw as-received
	Compressor running without fan	Control algorithm
	Tank thermistor failure	
	EXV failure	Gen 1 EXV faulty
	EWT thermistor failure	
	PRV leak (2 failures)	
	EXV failure	Gen 2 EXV
12Q503328 / Shari's Restaurant, OR	EVX failure	Gen 2 EXV
	Leak at discharge thermistor	
	PRV leak	
12Q503331 / Hilton, Glastonbury, CT	No persistent alarms	

Table 9.2: Detailed listing of field trial operational failures

EXV failures are a more critical issue as eight failures were observed. As discussed in Task 5, the valve supplier has been developing this valve. The first prototypes used in the EU HPWH prototypes and the NA phase 1 field trials were subject to extensive root cause analysis and found to fail because of internal vibration and fatigue at specific low flow rates. It is notable that four EXVs failed within a couple of months in the late winter of 2005-2006. The supplier tightened up some clearances to minimize vibration and added a spring washer for the next generation of valves. These valves were used originally in four of the NA phase 2 units and were also used to replace any of the earlier generation units that failed. This valve design also presented multiple failures in the NA field trials although surprisingly no failures have been reported by the EU field trials which could indicate dependence on the control algorithm. Two valves failed to operate in their respective HPWH unit but worked when returned to the supplier. A third suffered a failure of the new spring washer. No root cause investigations were completed on failures of the second generation of EXV and this is an outstanding reliability issue.

In-service water pump drive failures were experienced on two units. In addition, the Hilton unit (GHG) returned a pump drive failure alarm but the unit operated fine once the alarm was reset. No root cause investigation has been performed.

Sensors and wiring flaws accounted for a total of eleven failures. Sensors involved included the tank thermistor (3 failures), the entering water thermistor (3 failures), the compressor discharge and suction pressure transducers (1 each) and the high pressure switch (1 failure). No specific root cause analysis of these failures was performed although some units had observable problems as noted in Table 9.2. Wiring resulted in many early trips as units quickly shook loose inadequately tightened connections at the control box terminals, but these are treated as commissioning faults. In-service problems included a poor sensor connection and loosened 480V connections. These could also have been as-received conditions that did not fail immediately. In any case a procedure for assuring the quality of wiring connections is critical.

A number of leaks were observed. Two leaks occurred multiple times: six occurrences of leakage at the compressor discharge pressure relief valve (PRV) and two occurrences at the compressor discharge thermistor. Other leaks in the evaporator coil, service valve, compressor fittings, and service module components occurred only once. The PRV leaks have been observed in the EU units as well. Generally, the units leak after popping open, something that should not happen in normal operation. However, many of the valves have been observed to open at lower pressure than specified. It is also likely that pressure control overshoot resulted in closer than intended approach to the PRV setpoint, particularly when using earlier versions of the control software. We believe the combination of these effects resulted in the large numbers of PRV leaks. Improvements in the valve quality or software efficacy, or simply setting the control pressure setpoint lower, should reduce these failures.

Control algorithm issues caused a number of trips, primarily in the areas of setpoint overshoot, ancillary heater operation, defrost cycle initiation, and fan operation. These issues are expected given the nature of control algorithm development. All of these items were traced to algorithm errors and repaired.

Overall the operational failures were more frequent than expected but mostly of a nature that one might expect after reviewing Pareto charts for any new Carrier product. In any new product developed, control algorithms, wiring, and sensors cause early faults and experience tells us that Carrier's process is very efficient at fixing these issues. Leaks are also common in early development, although not as prevalent as seen in this very high pressure system. It is clear that a better qualified pressure relief valve is required. The other leaks may be substantially caused by shipping damage or manufacturing. Looking for signs of damage or manufacturing flaws that could lead to leaks are a prime focus of the unit teardown inspection.

The most critical reliability issue is clearly the EXV. Despite a total of six generations of vendor development this component is still not fully reliable.

Reliability growth model results

A reliability growth model was developed under Task 2 for ongoing prediction of upper and lower bounds of field failure rate of the overall system as well as high risk components if desired. As discussed in Task 5, the amount of useful data available from component testing is very limited, so only total system reliability is considered here. System data age all components simultaneously, so it is feasible to input individual component failure modes from the system data into the model and derive independent FFR estimates for each component modeled. However, given the small number of failures observed this exercise was deemed redundant. It is quite clear from the data that EXV failures, leaks and sensor failures dominate the system failure rate, while compressor or defrost valve failures do not.

To assess the current reliability estimate from the growth model in system mode the input requirements are straightforward. For each field trial unit one must input either the current unit accumulated hours or, if a failure has occurred, the unit hours at the time of failure. In addition the current estimate of unit run hours per year is input.

The reliability growth prediction as of October 2006, when the field trials as a group were effectively ended, is shown in Figure 9.2. The overall predicted system FFR estimate is still quite high. Although the phase 1 units had more failures, they also had much more run time, so their current FFR estimate is actually lower than that of the phase 2 units. At this time, the expected FFR for the phase 1 units is 73% at 80% confidence. Clearly, this work is incomplete and more field trial time is necessary to statistically assure reliable performance.

Task 10 – Create North American Product Specifications

The design for the CO2 heat pump water heater evolved during 2004 as we gained experience with both North American phase 1 field trials as well as the European field trials. Further, considerable component testing and modification was taking place on both programs leading to components of improved specification. Finally, where feasible, UL listed components were specified as discussed in Task 5. Overall, only a few changes for UL compliance could be incorporated into the specification in time to affect the phase 2 units. The specifications given here will include only these modifications unless otherwise noted.

It should be noted that new control algorithm versions were applied to all field trial units as they were developed, so control specification is not dependent on which phase the field trial unit belongs to. Also, some components were backfit to the phase 1 units as noted. Further component specification modifications motivated by UL compliance will be documented in the final specification in Task 14. The amendments to the original prototype specifications are as follows:

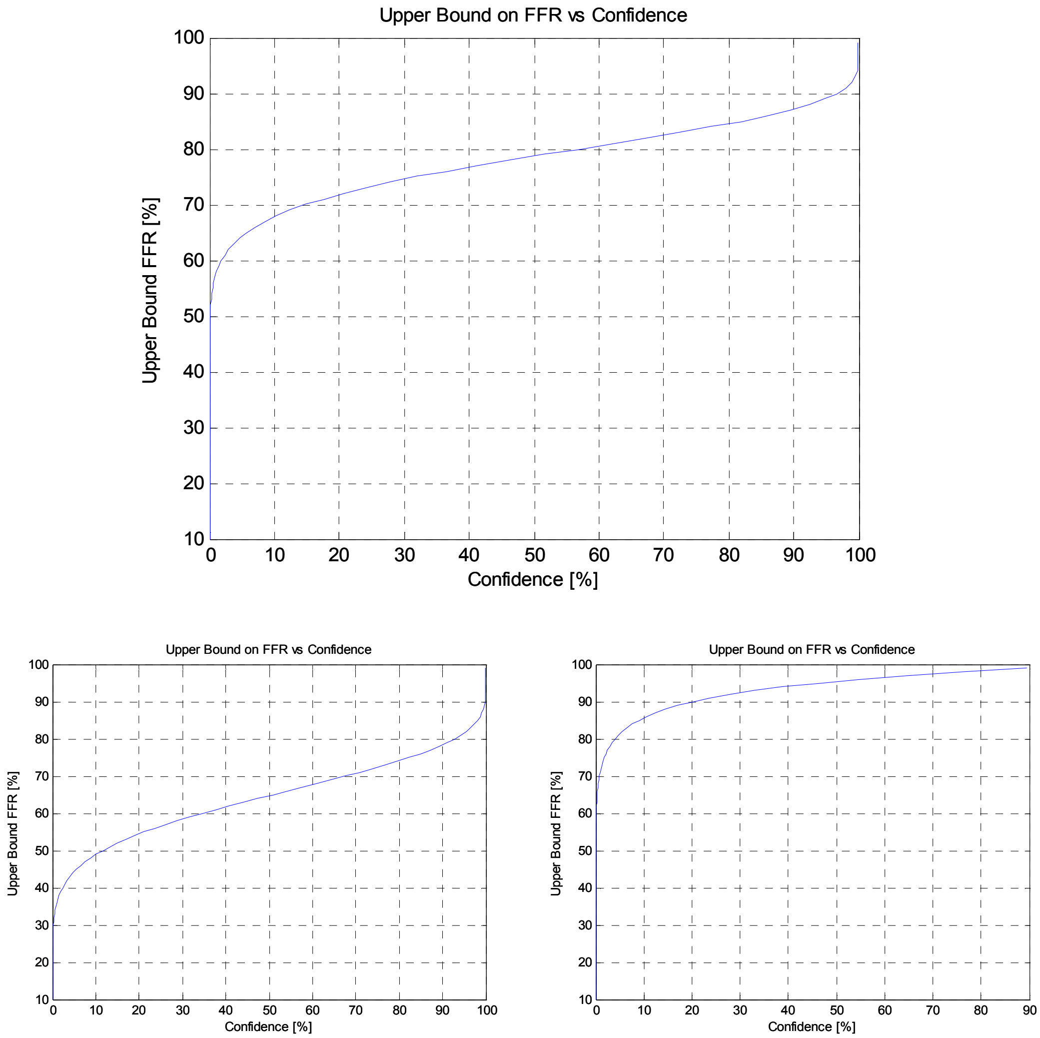


Figure 9.3: Reliability of all 8 FTUs (top), and of phase I and phase II FTUs (bottom)

- EXV: The vendor is continuing development of the EXV for this application. For the NA product specification we applied the vendor's fourth generation valve that has been modified to prevent damage caused by vibration.
- Service module: Although the specification for the service module has not changed from phase 1 to phase 2, it is worth noting that the supplier and manufacturing process has changed. In the phase 1 units the individual components were supplied to the Carrier plant and assembled there. For phase 2 the service modules are fabricated entirely by the new supplier, Danfoss. The only component in the service module not sourced from the supplier, the EXV, is shipped to the service module supplier for assembly. This change is expected to reduce the probability of leakage as well as reduce cost. It should be noted that the EXV manufacturer Saginomyia has since been acquired by Danfoss.
- Compressor oil cooler. As discussed in Task 5, compressor reliability testing has indicated that the combination of high heat generation and solubility of CO_2 under

certain operating conditions can significantly decrease the viscosity of the oil at high ambient temperatures. An oil cooler is expected to extend the life of the compressor bearings and aid motor cooling. Oil coolers have also been backfit to the phase 1 units

- Single speed UL listed fan: This design change minimizes any development costs/efforts associated with the fan through implementing an already UL listed fan that is currently used by Carrier in North America. The fan was also selected to resolve the issues of higher power draw and airflow associated with operating a 50 Hz design fan at 60 Hz.
- Electrical/control box: A detailed analysis with Siemens US has indicated that several design changes are necessary on the high-voltage section of the electrical/control box, to address component rating issues at high current draw conditions and to facilitate UL listing. A specification for this box has been prepared but the boxes could not be acquired in time to include them in the phase 2 field trial units.

Task 11 – North American Product Implementation (field trials)

Implementation of the phase 2 field trials followed a more rigorous process than the phase 1 units. A wide variety of sites were evaluated and prioritized, a process presented in more detail below. Seven phase 2 units built to the Task 10 specification were manufactured by the Carrier factory in Montluel. Six units were tested in a psychrometric room at UTRC to verify performance. Unlike the phase 1 units, the units were tested as-shipped, with no modifications to components or controls unless faults existed as listed in Task 9.

Five phase 2 units were placed in the field: two in the southeast, two in the northeast, and one in the northwest. The installation and performance of these units is reviewed in detail. A sixth phase 2 unit was maintained in a psychrometric room at UTRC for use in developing improvements and duplicating issues experienced in the field. Finally, the seventh unit was used to evaluate UL listed components and to support UL system tests as discussed in Task 5. The seventh unit was never fully functional and it was not tested for performance.

Phase 2 site selection

The process for site selection consisted of two phases: identification and inspection of potential field trial sites and merit-based downselection of the five best sites.

Site selection was a careful process, as the number of sites is limited, and we wished to derive the maximum value out of each of the sites. In general, locations were chosen to stress the operation of the units in a particular way, in order to allow for the most significant reliability information and failure modes to be developed. In addition, all sites required a backup source of hot water in the event that a failure occurs. We also considered locations that will highlight the energy saving and economic value potential of this technology. Finally, these locations were chosen to aid in disseminating general knowledge about this technology, which will address one of the key aspects of commercialization: pull from customers

UTRC leveraged Carrier national accounts, Carrier Marine Systems, and Carrier Commercial Service and Sales regional operations management. The latter identified a lead applications engineer in key territories including CT, western MA, Eastern NY, VT, Eastern NE, NYC, LI, NJ, PA, DE, MD, VA, Washington, DC, and southeast Texas. Sites of interest included a power utility, a power marketing energy service company (ESCO) in Boston with national reach, a naval contact to network into land based military sites, and DOE in West Virginia. Table 11.1 lists sites considered along with a few key characteristics of the application.

Site	Application	Projected Operational Savings	Position in Operating Envelope	Location	Trade Alley
Aramark, Syracuse	Commercial Laundry	7,396	High P ratio	outdoor	CCS
Coyne Textiles, Syracuse	Commercial Laundry	2,116	High P ratio	outdoor	CCS
Jail, Syracuse	Kitchen	10,635	High P ratio	outdoor	CCS
Sheraton, Syracuse	Hotel	4,729	High P ratio	out/indoor	CCS
Jail, Jamesville	Kitchen	1,756	High P ratio	outdoor	CCS
Nursing home Jamesville	Commercial Sanitary	1,137	Middle	out/indoor	CCS
Fairfield University	Dormitory/Cafeteria	4,721	High P ratio	outdoor	
Marriott Hotel, Farmington	Hotel	5,245	High P ratio	out/indoor	
NAVSEA Philadelphia	Cafeteria	2,437	Middle	outdoor	
Shari’s, Oregon	Restaurant	6,227	Middle	outdoor	PGN
Providence Health, Oregon	Boiler pre-heat	10,409	Middle	out/indoor	PGN
DOE Office Headquarters	Cafeteria	4,724	Middle	out/indoor	DOE
DOE Virginia	Cafeteria	2,742	Middle	out/indoor	DOE
CB Base, Gulfport	Cafeteria	7,793	High load	out/indoor	Southern
CSS Anheuser Bush	Industrial Sanitary	4,969	High P ratio	out/indoor	
Marriott, Atlanta	Hotel	Unknown			Southern
Grand Casino, Biloxi	Restaurant	4,429	High load	outdoor	Southern
Montevallo College	Dormitory	6,345		outdoor	Southern
Children’s Hospital, AL	Hospital	1,634	High load	out/indoor	Southern
Yeast USA, AL	Industrial Sanitary	13,243	High load	out/indoor	Southern
Prison, Nashville	Laundry	13,313	Middle	outdoor	TVA
Superior Catfish	Industrial Sanitary	7,343	Middle	Outdoor	TVA

Table 11.1: Field trial sites considered for phase 2

Carrier Commercial Services provided some feedback from their site interactions that apply to the US market for this technology.

- Hospitals use electric heat to make steam for sterilization, autoclaves, etc. and there is a good fit for sanitary water pre-heaters.
- Justice Centers (aka jails or penal detention centers) such as the one under consideration in NY requires a steady and substantial volume of hot water, not just for the 750 hand sinks but also for the kitchen and the laundry. This facility has the storage capacity of 1500 gallons of hot water and consumption greater then any other building in the county complex.
- Everyone likes the idea of highly efficient electric heat for hot water.

- There is a lot of constructive and positive enthusiasm for seeing this technology make it to commercialization.
- Commercial laundries are looking for a solution in the 50, 100, 150 GPM range (10x, 20x, & 30x) providing either multiple unit orders or justification for future development to reach new markets for impact.
- Outdoor installations are often convenient for the temporary field trials in North America. Customers seem agreeable to permanent indoor installations as well. A second order value from free cooling is deemed valuable and worth incorporating into value propositions as a differentiator over conventional electric hot water heaters.
- Commercial laundries see the indoor installation of CO_2 Heat Pump as a free cooling method of improving the condition of the work place – leading to employee retention. Employee turn over is apparently quite high in this vertical market.
- Commercial kitchens see the free cooling as a way of providing a level of comfort it would not otherwise provide because the cooled air is constantly exhausted via range hood exhaust hoods.

UTRC developed a scoring methodology aimed at impartially rating the worth and merit of each candidate field trial site to the program. Cost, projected run hours, customer enthusiasm, extrinsic value of the networking trade ally (potential technology advocate / influencer) and half a dozen other parameters were used to score the candidate sites within a "portfolio" style scoring environment in which program members openly discuss the merits and risks associated with each potential host site. The general figures of merit are shown in Table 11.2.

Field trial selection criteria		
1	Economic	Yearly savings
2	Application	How much is the unit stressed? More is better.
3	Application	How different is this application from others we currently have?
4	User	How supportive / proactive is the field trial site owner?
5	Marketing	How likely is this field trail going to attract future sales?

Table 11.2: Field trial selection criteria

The potential sites for the Phase II FTUs were then narrowed to five, taking into account factors such as the annual operational savings, projected operating hours, diversity value, user enthusiasm, marketing potential, installation/monitoring costs, significance of application, and the installation location. The last site was originally planned to be a Naval Base on the gulf coast, but this site was damaged in the summer of 2005 by Hurricane Katrina. A last minute opportunity to place a unit at a large hotel in Connecticut was judged to be the beat alternative. These sites are summarized in Table 11.3 below.

Site	Annual Run Time [hr/yr]	Projected Annual Savings
Fairfield University, CT	6552	$ 4,721
Superior Fish Products, MS	4380	$ 9,168
Eastern Health Systems, AL	4368	$ 8,556
Shari's Restaurant, Oregon City, OR	5824	$ 6,227
Hilton Garden Inn, Glastonbury, CT	7280	$5,245

Table 11.3: Final phase 2 field trial sites with anticipated operational data

Field trial installation

After evaluating all potential sites, the primary drivers in the choice of phase 2 field trial sites were seen to be diversity of application and the level of interest displayed by the site partners. Diverse climate (indoor or outdoor), high potential operating hours and savings were also significant contributors to the final outcome.

The first phase 2 field trial unit was installed at Superior Fish Products in Macon Mississippi. This site was nominated by TVA, a very enthusiastic proponent for HPWH technology. Superior Fish uses hot water to clean machinery used to process fish. The expected hot water usage was 2880 gallons per day. The location is also notable for the highest potential cost savings of the phase 2 trials primarily because of the electric-gas price differential. This unit, shown in Figure 11.1, was placed inside an unoccupied storage room which had relatively little need for cooling. The staff at Superior Fish performed the installation, which resulted in the lowest cost of all installations, only $2,829. The Superior Fish unit was commissioned in October 2005 and was decommissioned in January 2007. In this time the unit operated for 3735 hours and experienced 7291 compressor starts.

The second phase 2 unit was installed at Fairfield University in Fairfield Connecticut. This site was nominated by Carrier Commercial Services (CCS), who also performed the installation. The Fairfield unit was installed in a gymnasium, supplying hot water to the shower facilities. It provided preheating to a steam/water heat exchanger fed by facility steam and was expected to supply 4308 gallons per day. This installation, shown in Figure 11.2, is notable for the separation between unit and tank and the use of tandem tanks. The CO_2 HPWH unit was placed outside separated from the tank inside by about 35 feet of pipe including 24 elbows. Two storage tanks were installed in series to increase capacity while maintaining a single control point in the bottom tank. This type of installation results in less stratified cold water in the tank than would result if two tanks were placed in parallel. The installation by CCS, shown in Figure 11.3, had an installation cost of $6,509. The Fairfield unit was commissioned in November 2005 and was decommissioned in January of 2007. In this time the unit operated 1203 hours and experienced 4053 compressor starts. Anecdotally, the Fairfield staff informed us that during the winter of 2005-6, when facility steam suffered an unplanned shutdown, the gym provided the only hot showers on campus.

The third unit was installed in a medical center operated by Eastern Health Services in Birmingham Alabama. The site was one of three successfully nominated by Southern Company,

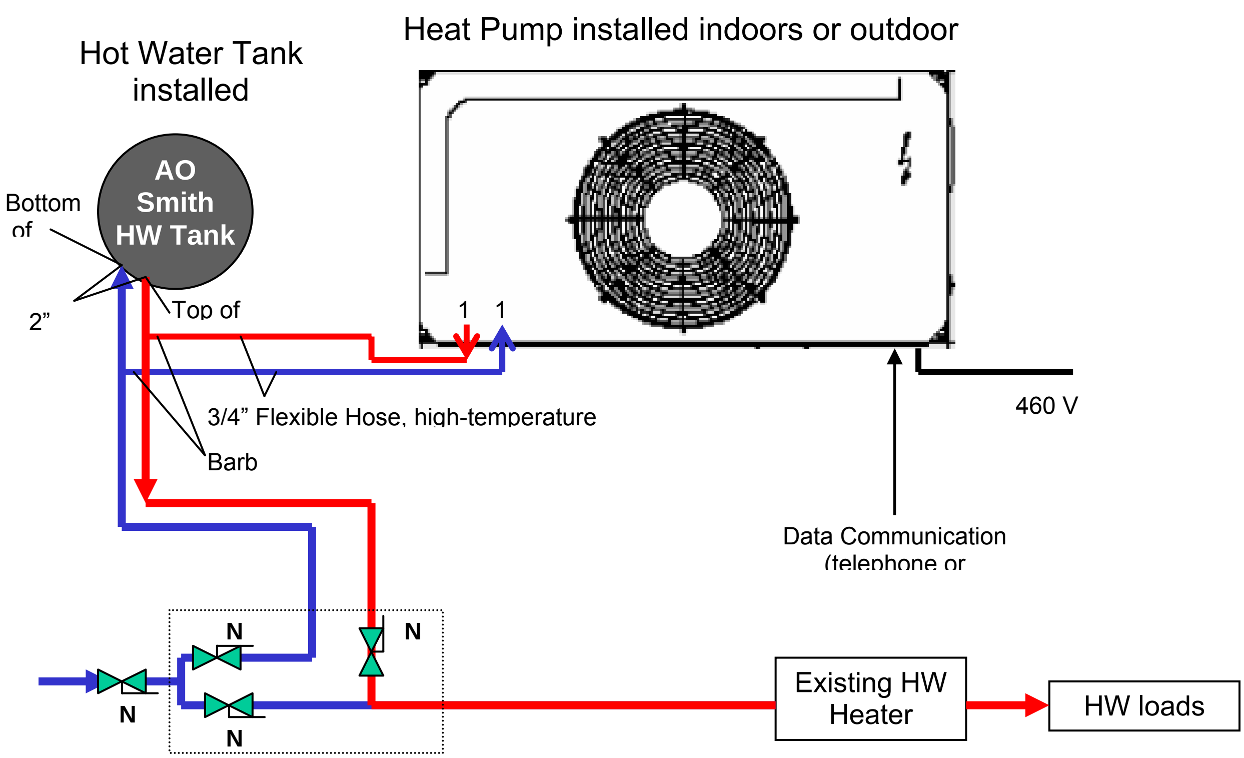


Figure 11.1: Superior Fish Installation

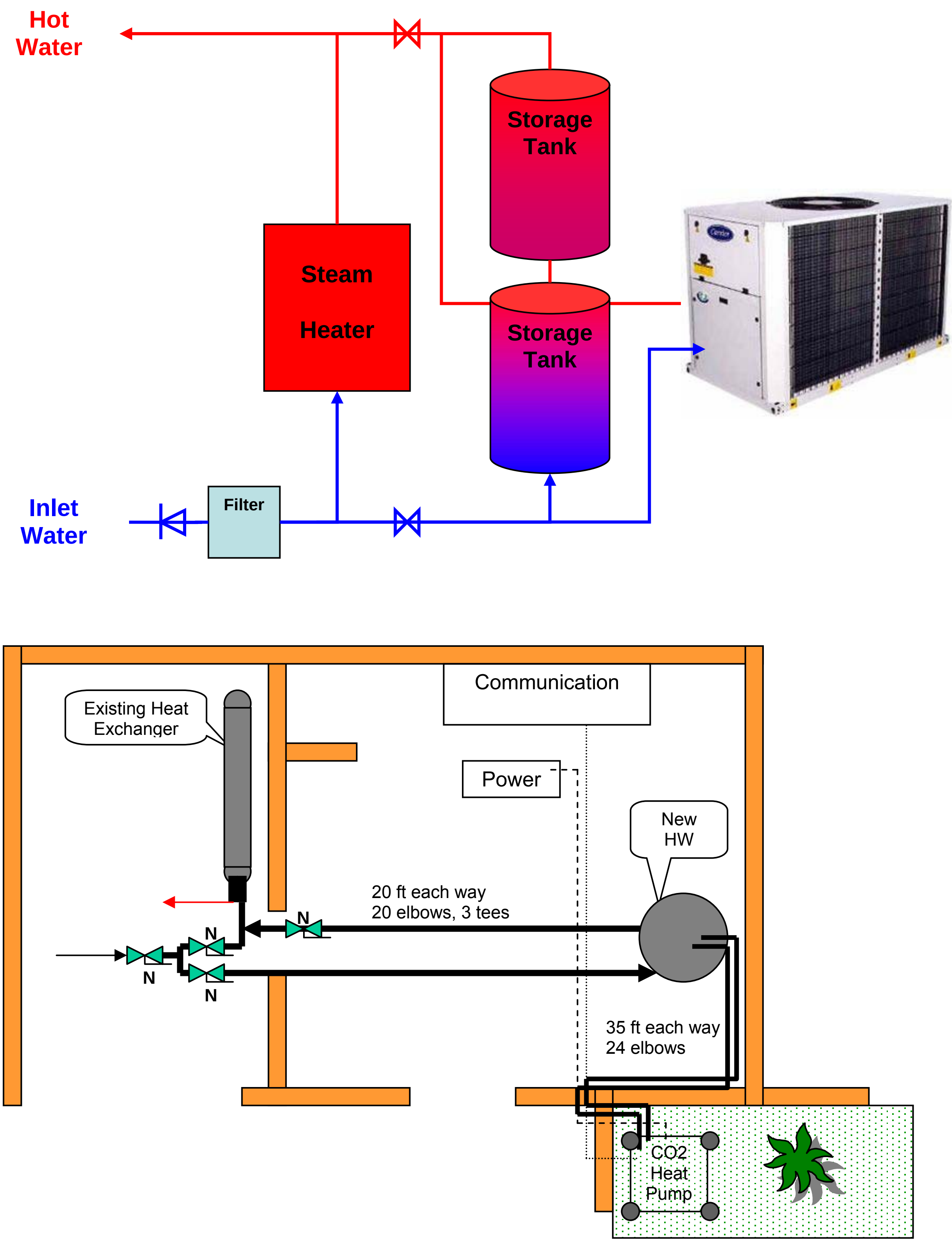


Figure 11.2: Fairfield University installation diagrams showing twin tanks

Figure 11.3: Fairfield University Components

who strongly support the development of HPWH technology. The unit supplies general hot water to the center and was installed as a preheater to the existing facility hot water heater that could be valved in or out as desired. The expected demand for the site was 2872 gallons per day. The unit was installed inside a mechanical room near the center of the building as shown in Figure 11.4 and provides cooling to offset the heat generated in the room. The unit was installed by Midsouth Controls at a cost of $5,850. The unit was also commissioned in November 2005 and was decommissioned in November 2006. In this time the unit operated 1638 hours and experienced 3951 compressor starts.

The fourth unit was installed in a Shari's Restaurant in Portland Oregon, part of a large chain of restaurants. This site was nominated by Portland General Electric, one of several electric utilities that promote HWPH technology, and represents a very large potential commercial market application. The unit supplies general hot water to the restaurant with an expected usage of 3829 gallons per day, and was installed as a preheater to an existing water heater. The installation, shown in Figure 11.5, is notable in that the HPWH unit and tank were installed on the roof of the restaurant when no suitable ground level site was found, and also in the requirement for a 208/480V step-up transformer. Because of this the installation, performed by Carrier Commercial Services with support from Accurate Heating, was certainly the most expensive at $20,150, but Energy Trust of Oregon contributed $7,000 to the cost of installation. The Shari's unit was commissioned in December 2005 after extended permitting including seismic analysis, and was decommissioned in January 2007. In this time the unit operated 1278 hours while experiencing only 1118 compressor starts, indicating good hot water demand when the unit was operational.

Figure 11.4: Mechanical room installation at Eastern Health Services

Figure 11.5: Rooftop installation at Shari's Restaurant in Portland Oregon

The final phase 2 field trial unit was installed at a Hilton Garden Inn hotel in Glastonbury Connecticut. This trial site was a last minute selection, identified by UTRC when Hurricane Katrina leveled the originally planned naval base site. We had been pursuing a hotel application because of the large potential market they represent, but had not previously found a suitable site. The unit supplies general hot water to the hotel and is installed as a preheater to the existing water heater. It is a split installation with the HPWH outside and the tank placed in a laundry room, although not separated by as large a distance as the Fairfield installation. The expected usage of this site was predicted to be the highest of all trials at 4787 gallons per day, although the total cost savings was predicted to be lower than average. The unit was installed by Crest Mechanical, the incumbent facility mechanical contractor, at a relatively high cost of $10,750. The Hilton unit was commissioned in February 2006 and decommissioned in January 2007. At this late stage of the field trials our resources became very limited and we were never able to establish remote communication with the unit. With little on-site staff available to verify operation, we were not able to easily learn when the unit had tripped. Because of this the unit only operated 247 hours with 754 compressor starts even though it had no hardware failures.

A sixth field trial unit was completed and tested in a psychrometric room at UTRC. It was maintained there as a means to test control algorithm or component upgrades and to simulate field failures if necessary. It was not installed in the field and received few operational hours.

Table 11.4 summarizes the installation and operational data for the five phase 2 field trial units. These units accumulated 8101 operational hours over 64 unit-months. Again, a variety of contractors installed the units based on the recommendations of the field trial participants. Superior Fish used in-house facilities staff for the installation, resulting in the lowest cost of any field trial unit, while the other sites employed outside contractors resulting in higher costs. The high cost for the Shari's installation represents the rooftop installation and resulting permitting. Carrier Commercial Services performed the Fairfield University installation as part of a larger renovation of University services, and also contracted the Shari's installation.

Site	Compressor Run Hours	Compressor Starts	Contractor	Cost of installation
12Q503329 (SCF) Superior Fish, MS	3735	7291	Superior Fish	$2,829
12Q503332 (FFU) Fairfield University, CT	1203	4053	Carrier Commercial Services	$6,509
12Q503327 (EHS) Eastern Health Services, AL	1638	3951	Midsouth Controls	$5,850
12Q503328 (PGN) Shari's Restaurant, OR	1278	1118	CCS / Accurate Heating	$20,188
12Q503331 (GHG) Glastonbury Hilton, CT	247	754	Crest Mechanical	$10,750

Table 11.4: Installation and operational summary of the phase 2 field trial units

Field trial performance

Averaged performance data are compiled in Table 11.5 for the five phase 2 units under varying ambient conditions and compared to the design performance originally presented in Figure 4.1 in Task 4. As was the case for the phase 1 field trial units, it is clear that the observed performance is less than the design performance typically by 20-30%. The Shari's unit showed very poor performance, but, like the DCH phase 1 unit and as discussed in Task 9, we later learned that the water flow meter was calibrated to read only 50% of the actual flow. Correcting for this brings the performance in line with the other units. As discussed in Task 7, the COP data are low by 20-30% because of leakage resulting from poor compliance with the valve plate specification. Only the SCF unit appears to be consistently below average. Although not proven, the SCF compressor teardown discussed in Task 13 provides some insight into this deviation.

Site	Ambient air temperature OAT (ºC)	Entering water temperature EWT (ºC)	Leaving water temperature LWT (ºC)	COP	Design COP (deviation)
Superior Fish	4	23	58	2.0	2.6 (-30%)
(inside unheated)	10	24	64	2.3	3.0 (-30%)
	17	25	62	2.7	3.3 (-22%)
	24	25	62	2.8	3.6 (-28%)
Fairfield University	-10	9	61	1.8	2.4 (-33%)
(outside)	-2	10	61	2.2	2.6 (-18%)
	2	9	61	2.4	3.0 (-25%)
	7	8	61	2.7	3.2 (-18%)
	20	22	61	2.9	3.4 (-17%)
	31	28	61	2.6	3.7 (-42%)
Eastern Health Services	20	12	61	3.1	3.8 (-22%)
(inside)	24	12	62	3.3	4.0 (-21%)
Shari's Restaurant *(outside roof)*	12	15	62	1.4 (2.8)	3.3 (-18%)
Hilton Garden Inn *(outside)*	12	30	61	2.5	2.9 (-16%)

Table 11.5: Envelope performance of the five phase 2 field trial units

Task 12 – Develop a Sizing, Specifying, Cost Tool

The intent of this task was to develop a sizing, specifying, and cost tool that would assist sales outlets and end customers in understanding the product, its best application/installation, and how to appropriately size and select this product for their individual needs will be developed. This tool was to provide projected estimates for the cost of ownership, installation, maintenance, and operation of a CO2 HPWH product, explaining payback periods and any potential free cooling benefits. Because of limited resource availability in 2006, we set the highest priority on data collection, so the full function of this tool has not been realized. The tool to date does perform the most difficult part of specification, optimization of the number of HPWH units and tank capacity needed to achieve the lowest initial and operating cost for given customer requirements.

The tool has been implemented as a Microsoft Excel application. This platform offers the opportunity of easy integration with report-generation software and portability to PDA equipment for use by sales representatives in the field. Currently the tool offers the user a selection of three applications: a motel, restaurant, or laundry. Hourly profiles for these applications are stored in the tool. Customer requirements for water consumption, tank capacity, and reserve are input, as are local electric costs, initial unit HPWH cost and tank specific cost. The tool then optimizes the number of HPWH units and tanks for the specific set of requirements.

The tool, provided separately, is functional but incomplete. Other data derived for use in the tool but not yet incorporated are also provided in separate tabs of the attached spreadsheet tool. These include maximum and average hot water usage for eight applications developed from the ASHRAE applications handbook, ambient temperature bin hour data for 50 US cities, and a format for installation and maintenance costs. A flowsheet of the planned tool functionality, intended to be a user's guide, is also included.

Task 13 – Tear-Down and Inspection of Units

Each of the eight field trial units were destructively examined with the intent of determining rates of component wear and identifying any other issues such as physical damage or improper manufacturing technique. Budget restrictions limited the number of detailed measurements that could be taken, so our efforts focused on critical areas such as leaks and compressor wear.

Units were decommissioned at each site by a local contractor, in all cases a contractor who had been involved in installation of units, although not necessarily those who had installed that specific unit. By the time decommissioning was begun most of the units were in alarm, and most of these were indicating leaks or an EXV failure. The units were put on a pallet, wrapped in plastic, and trucked to UTRC where they were placed in a storage area awaiting inspection.

The units were first pressurized to 1000 psig, the highest pressure reasonably allowable for the whole system, and checked for ability to hold pressure. If any leakdown was noted the unit was charged with a small amount of R-22 and a leak detector was used to spot leak locations. One lesson learned is that the braze-plate gas coolers are not able to withstand freezing even after the hydronic system is opened and drained of water. Only the GKN and UTRC phase 1 units, and the EHS and SCF phase 2 units, could be pressure tested: the others all leaked from refrigerant to water side. No freezing was experienced with any of the units in service; all failed at UTRC awaiting inspection. Of the four units that could be checked for leaks, only the GKN unit would hold pressure. The other three had identifiable leaks tabulated below.

Once leak-checked a visual inspection of the case is performed. All units had acceptable compressor oil level. Two units showed small oil leaks, usually an indicator of a refrigerant leak. No unexpected debris was found in any unit. Next, all electrical connections were removed. In one phase 2 unit a failed connector was identified. Then all tube fittings were separated and tubing cuts were made to remove each section of tubing. Fittings and tubing were examined for any abnormalities. In all cases tubing was found to be clear from oil and debris, and no tubing damage was found. In several cases mild scale was found on the hydronic circuit tubing. All fittings were undamaged, but in one phase 1 unit the fitting at the oil cooler / compressor pump housing connection was found to be improperly manufactured. The tubing was cut too short so the fitting ferrule was located right at the end of the tube. This fitting was not found to be leaking during pressure testing. Each system component was then removed and examined for apparent damage. Apart from indications of scaling on gas coolers, no manufacturing or shipping damage was noted.

Only the compressors were subject to further inspection, although sealed components such as the EXVs and defrost valves were saved for potential future destructive inspection. The compressors were fully disassembled. Measurements of bearing diameters, piston and bore diameters, and ring gaps and weights were performed and documented. Main, rod and wrist pin bearings were all examined for visual indications of damage. In all cases rod bearings were found to be scratched by particulates in the oil. This type of damage had been widely experienced while trying to qualify the EU compressors, and specification of an oil filter was necessary to satisfactorily resolve the problem even after repeated efforts to improve manufacturing cleanliness. No material transfer was observed. Only the UTRC compressor was found to have excessive debris on the oil sump magnet, most likely resulting from the compressor failure experienced by this unit. One wrist pin was found to be a tight fit into the rod and the bushing was noted to be scuffed on one side, an indication that the bushing was out of round.

More interestingly, the SCF unit compressor internal components, including bearings, pistons, cylinder bores, and valves, were found to be coated with copper. Small chunks of copper were found imbedded in the rod bearings and laying on the bottom of the oil sump. No source for this copper was identified in the compressor, so our hypothesis is that cutting and brazing operations to repair the EXV released copper into the system. The filter/dryer, which might have trapped such particles, is located upstream of the EXV specifically to protect the valve.

Valve plates were not examined for flatness given that poor flatness was endemic to this series of plates, but they will be saved for potential future examination.

Table 13.1 gives a summary of the findings of the system inspections described above. Figures 13.1 and 13.2 show the results of wear measurements. In Figure 13.1, the loss of piston ring mass from the eight field trial compressors is shown compared to similar data observed in EU prototype compressor testing. The uncertainty on the measurements is 0.075% but the original ring weight was not recorded so manufacturing scatter cannot be identified. The ring wear follows the general trend of the previous data except in the case of the GKN and UTRC units, identified by their total operational hours. No analysis has been performed to determine root cause, but the presence of excessive ferritic wear material on the sump magnet of the UTRC unit compressor was noted in Table 13.1. It is likely that wear in this compressor increased as the result of circulating debris released into the system by the compressor rod bolt failure early in life.

Figure 13.2 shows the rod journal bearing diametral gap observed in the eight field trial units compared to the as-manufactured specification. Although many of the gap measurements exceed the specification, this is not unreasonable for normal wear. Also, the original gap was not measured, so it is not known if the gap originally met specification. What is clear is that there is no observable trend with compressor hours.

The compressor inspection shows no consistent wear or failure mechanism that is expected to impact compressor reliability with the exception of the bearing surface scratching resulting from circulating debris. An oil filter will be added to the final specification to address this issue. This approach has been verified in EU compressor testing.

Unit	Failure	Hypothesis
12Z318130 / GKN GKN Aerospace, AL	Improperly fitted ferule on oil pump outlet tube	Manufacturing error
	Wrist pin tight, bushing worn off-center	Rod drilled out-of-round
	Rod bearings scratched	Manufacturing / wear debris in oil
12Z318131 / UTRC UTRC Cafeteria, CT	Leak at fitting upstream of oil cooler CO2 inlet	Fitting inadequate for this application
	Excessive debris on compressor sump magnet	Debris from early compressor failure
	Rod bearings scratched	Manufacturing / wear debris in oil
12Z318132 / DCH Druid County Hospital, AL	Gas cooler frozen after decommissioning	Stored improperly before teardown
	Rod bearings scratched	Manufacturing / wear debris in oil
12Q503329 / SCF Superior Fish, MS	Copper particles in compressor	Debris from EXV service or oil cooler
	Leak at charge port cap	Service valve worn or damaged
	Rod bearings scratched	Manufacturing / wear debris in oil
12Q503332 / FFU Fairfield University, CT	Gas cooler frozen after decommissioning	Stored improperly before teardown
	Rod bearings scratched	Manufacturing / wear debris in oil
12Q503327 / EHS Eastern Health Services, AL	Compressor discharge sensor damaged	Damaged by service personnel
	Leak at service module fitting	Incorrect fitting – SAE-metric adaptor
	Leak at compressor oil sight glass	O-ring pinched in manufacturing
	Rod bearings scratched	Manufacturing / wear debris in oil
12Q503328 / PGN Shari's Restaurant, OR	Gas cooler frozen after decommissioning	Stored improperly before teardown
	Rod bearings scratched	Manufacturing / wear debris in oil
12Q503331 / GHG Hilton Garden Inn, CT	Gas cooler frozen after decommissioning	Stored improperly before teardown
	Rod bearings scratched	Manufacturing / wear debris in oil

Table 13.1: Findings from field trial unit destructive examination

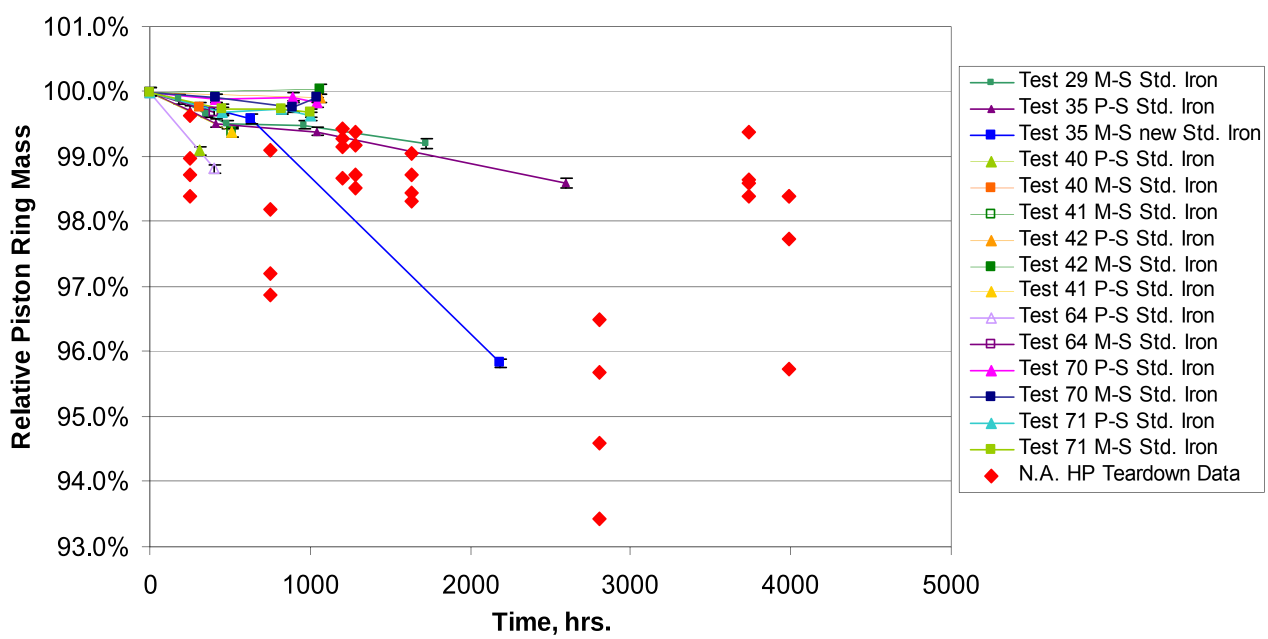


Figure 13.1: Piston ring wear data from the NA field trials compared to compressor test results

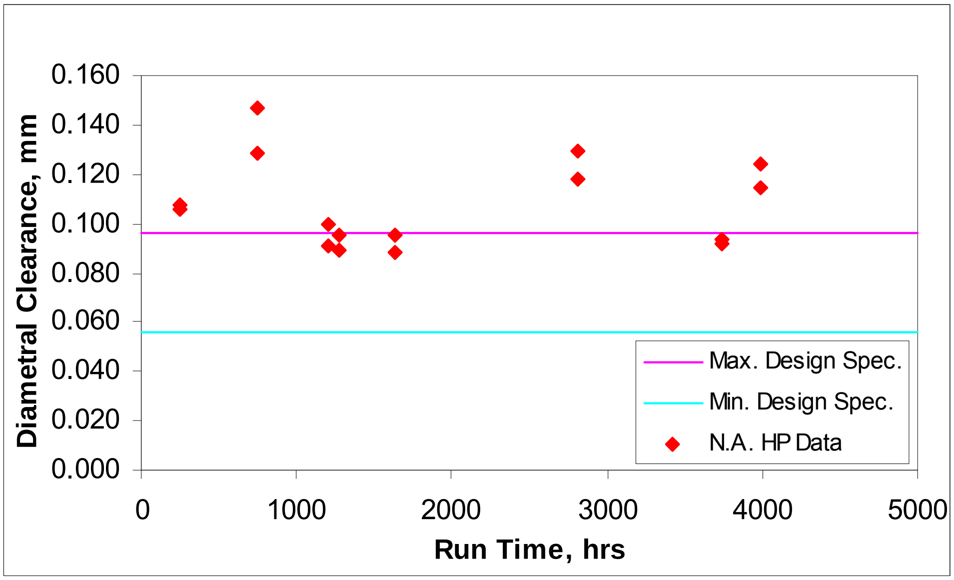


Figure 13.2: Rod journal bearing clearance data compared to new component specification

Task 14 – Redesign Based on Lessons Learned

A number of lessons have been learned from the NA field trials as well as from the EU field trials and EU compressor qualification during the duration of the DOE contract. The general specification appears to be well suited to the application, so design changes proposed here are primarily small changes in hardware, in hardware quality, and in commissioning procedure.

The one outstanding reliability element is the EXV. This component was initially suffering from high cycle fatigue. Changes implemented fixed the initial problem but part of the fix, a spring washer, subsequently presented fatigue failures. In addition, several valves clearly jammed in service but operated properly once removed. It seems most likely that broken spring washers were responsible but this was not verified. The manufacturer noted light scratches on a rotating component that may give some clues to the problem, but no resolution has been tested. It is clear that this component requires substantial further development. However, the technology for this valve appears adequate and we expect no inventions are needed to achieve success.

A number of hardware and procedural changes are recommended below to address reliability and performance each in order of priority.

- Replace the high pressure relief valve with a burst disk. The pressure setting should be raised to 2200 psia, so pressure testing requirements for UL listing must also be revised.
- Remove all possible fittings and replace with brazed connections. Although not proven, we believe that braze fittings fail in manufacturing or shipping rather than in service. Fittings at the oil cooler as well as connectors and adapters in the service module could be brazed although new components would be required.
- Fit an oil filter at the oil cooler return upstream of the compressor. A filter specification of 3 microns with a beta > 75 appeared adequate for EU compressor qualification testing. The unit should have braze fittings, a burst pressure > 3900 psig, and flow / pressure drop

ratings appropriate for the compressor oil pump. Parker has prepared a prototype filter based on their 15P-1 element in a heavy braze fitting envelope.

- Fit protective shields on the transducers and connectors exposed to service personnel. All hydronic-side thermistors including the tank thermistor as well as the compressor discharge pressure sensors are vulnerable.
- Apply insulation to the evaporator capillaries. The capillaries may vibrate in service or shipping. Heavy insulation will damp vibrations and reduce braze vulnerability.
- Provide a trap between the EXV and accumulator to ensure debris from servicing cannot reach the compressor. Once service procedures are developed this should take the form of an up-pipe downstream of the service joint downstream of the EXV. This could be at the EXV outlet or at the service module outlet.
- Perform a leak check for each unit upon commissioning.
- Retighten wiring upon commissioning (or specify better wire connections at the controls).
- Consider water treatment if water is too hard or scale is known to exist on current water heating hardware.

Several other changes are recommended for improving performance.

- Maintain quality on valve plate flatness. A typical refrigerant compressor flatness specification is adequate, but it is critical for performance that this specification be reliably met.
- Reduce the oil cooler capacity. The oil cooler used in the NA and second generation EU units was a coiled tube-in-tube unit that has more capacity and more cost than necessary. A simple air cooled coil or smaller refrigerant-oil cooler is sufficient.
- Size the gas cooler to the application. Doubling the UA of the gas cooler specified for the HPWH results in a 10% increase in efficiency at the design point. This heat exchanger is expensive, so specifying a range of gas cooler capacities would allow a price/performance optimization to meet a particular requirement. This is a feature that could be included in the specifying tool.

Final specification and qualified suppliers

The CO2 HPWH is a heat pump water heater that utilizes CO2 as a refrigerant in a transcritical cycle to produce hot potable water. The general product specifications are as follows:

- Outdoor installation and use.
- Ambient storage temperature: –40 °F to 140 °F.
- Operating ambient temperature: –4 °F to 115 °F.
- Inlet water temperature (steady-state operation): 33 °F to 95 °F.
- Outlet water temperature (steady-state operation): 102 °F to 176 °F.

- Ambient humidity: 0% to 100%.
- Maximum allowable refrigerant low-side pressure: 1200 psig.
- Maximum allowable refrigerant high-side pressure: 2000 psig.
- 480 VAC, 3-phase, 60 Hz power input, 40 Amps max continuous draw.
- All refrigerant-side components must have construction material compatible with CO2 and PAG oil.

The major components of the system include:

- Chassis – physical structure with access panels.
- Control Box Module – enclosure with all the necessary electronics and control circuits.
- Compression Module – compressor and ancillary components for safety and control.
- Gas Cooler – double-walled for sanitary applications.
- Service Module – EXV, filter/dryer, defrost valve, and service valves.
- Evaporator – distributor, coil, and outlet header.
- Accumulator.
- Fan Module – fan blades, motor, and support structure.
- Hydronic Module – water pump and ancillary components for freeze protection, control, and serviceability.

The system and all its components must be UL recognized and listed, while meeting the required ambient storage conditions, operating ambient conditions, and maximum allowable refrigerant pressures where appropriate. This requires the refrigerant side components to meet a minimum burst pressure of three times the maximum operating pressure.

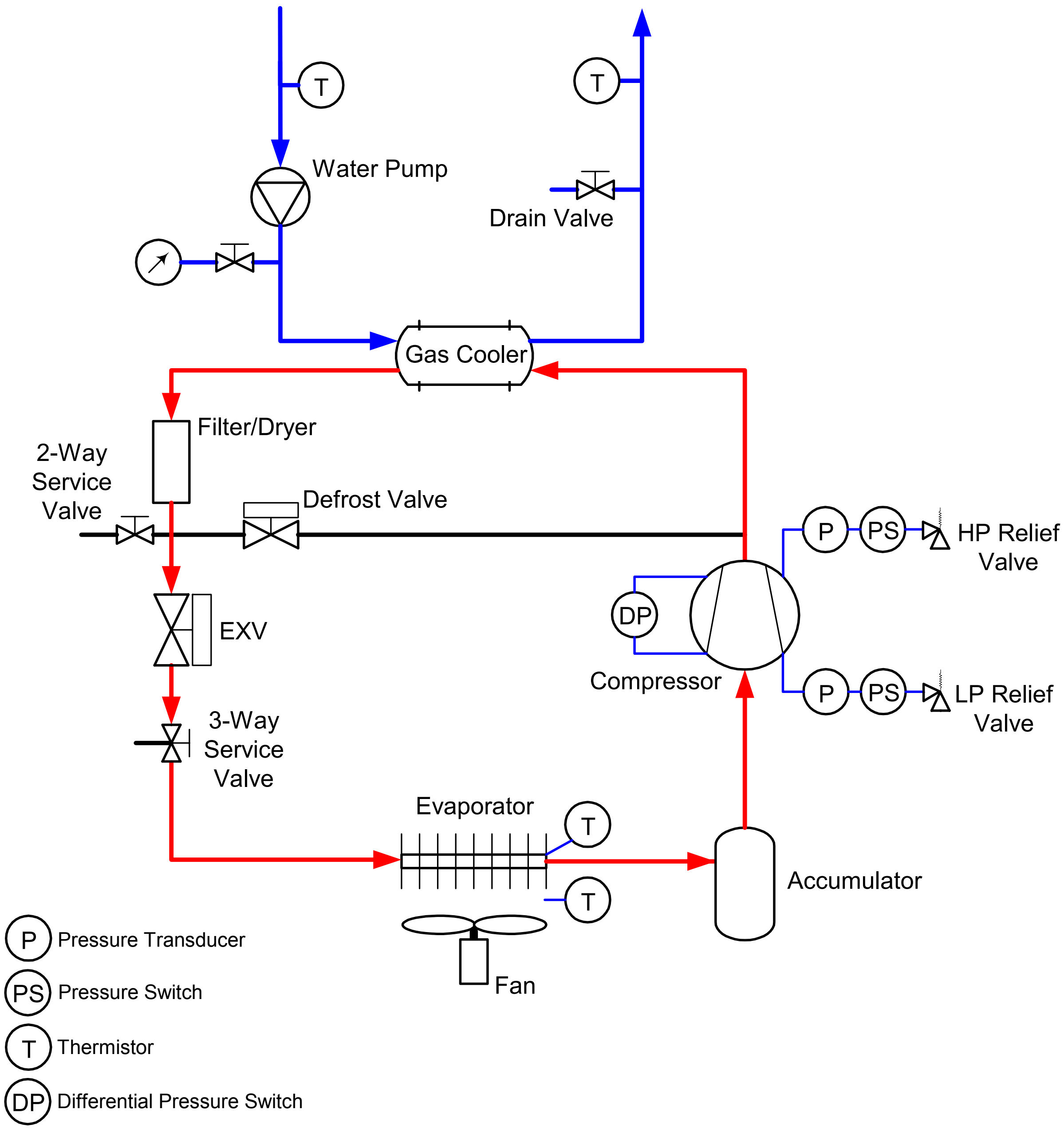
- Refrigerant low-side minimum burst pressure: 3600 psig.
- Refrigerant high-side minimum burst pressure: 6000 psig.

In addition, the hydronic module and the gas cooler should meet all local sanitary and potable water codes and regulations.

<u>CONTROL BOX MODULE</u> *(supplier: Siemens)*

The control box module contains the electronics necessary for unit operation. The primary components include power distribution equipment to the necessary points of use, the contactors, circuit breakers, and fuses necessary to safely operate all the electrical components, the control board for inputs, outputs, and the logic to control and operate the unit, and the user interface panel, which provides a means for accessing system information and set-points.

System Schematic Diagram



Installation and Connections

The control box itself should be a self-contained enclosure capable of protecting the electrical components from any weather damage. The control box should be divided into a high voltage and low voltage section, each with separate access doors. The low voltage section should contain the control board and user interface panel, which is accessible even when the unit is in operation. Main power to the unit should be connected to the high voltage side of the control box and locked out when main power is turned on to the unit.

Design and Operational Constraints

The control box module should be designed for outdoor and storage conditions.

Technical Characteristics

The control box should have the following characteristics:

- 480 VAC, 3-phase, 60 Hz power input.
- 40 Amps max continuous draw.
- 480 VAC to 24 VAC transformer for low voltage loads.
- Designed for the compressor, fan, water pump, and control box maximum loads.

COMPRESSION MODULE – COMPRESSOR *(supplier: Dorin)*

The function of compressor is to compress refrigerant vapor from low pressure to high pressure.

Installation and Connections

The compressor connects the outlet of the accumulator to the refrigerant side inlet of the gas cooler, as indicated in the System Schematic Diagram. The compressor body should contain ports to accommodate the ancillary compression module components, which are the low and high side pressure transducers, pressure relief valves, and pressure switches.

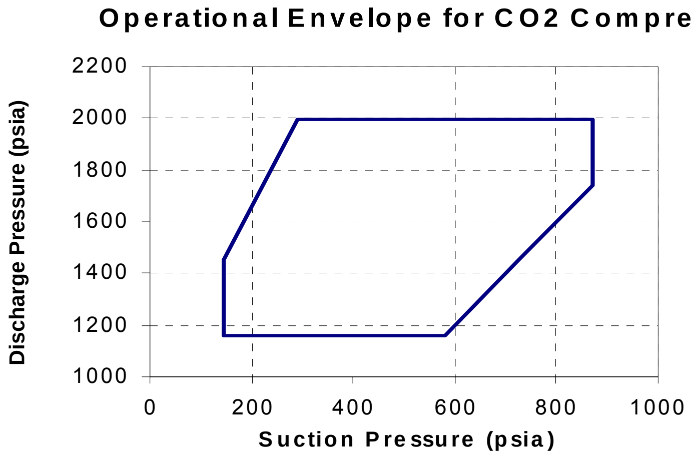
Design and Operational Constraints

In addition to the outdoor and storage conditions, the compressor is subject to the following refrigerant side constraints:

- Suction side: meets refrigerant low-side max operating and min burst pressure.
- Suction side: refrigerant temperature range: -40 °F to +115 °F.
- Discharge side: meets refrigerant high-side max operating and min burst pressure.
- Discharge side: refrigerant temperature range: +32 °F to +350 °F.

Technical Characteristics

The compressor must be capable of steady-state operation within the boundaries of suction and discharge pressures defined in the figure below. Transients such as startup and defrost will require brief operation beyond the steady-state operational envelope.



COMPRESSION MODULE – ANCILLARY *(suppliers: TI, Parker, Hoerbigger)*

The ancillary components of the compression module are the low and high side pressure transducers, pressure switches, pressure relief valve and rupture disc, oil differential pressure switch, sump heater, oil cooler, and oil filter. The pressure transducers provide information to the control system for startup, COP optimization, defrost, and safety. The pressure switches provide a means of shutting off the compressor in the event of excessive pressure on the high-pressure side, or too low pressure on the low-pressure side. The pressure relief valves provide a means of refrigerant vent in abnormal cases of overpressure in the system. The oil differential pressure switch provides a means of detecting sufficient oil pressure for compressor lubrication. The sump heater provides heat to the compressor when it is not in operation.

Installation and Connections

All ancillary components should be mounted directly to the compressor, as indicated in System Schematic Diagram.

Design and Operational Constraints

In addition to the outdoor and storage conditions, the ancillary components are subject to the following refrigerant side conditions:

Low-pressure side pressure transducer, switch, relief valve, and oil differential pressure switch

- Suction side: meets refrigerant low-side max operating and min burst pressure.
- Suction side: refrigerant temperature range: -40 °F to +120 °F.

High-pressure side pressure transducer, switch, and relief valve

- Discharge side: meets refrigerant high-side max operating and min burst pressure.
- Discharge side: refrigerant temperature range: +32 °F to +350 °F.

Oil filter

- Compatible with refrigerant and oil
- Meets appropriate pressure drop requirements
- Meets appropriate oil temperature requirements

GAS COOLER *(supplier: SWEP)*

The function of the gas cooler is to heat exchange hot refrigerant to cold water. It serves the same purpose as a water-to-refrigerant condenser in a conventional heat pump water heater.

Installation and Connections

The gas cooler connects the inlet/outlet water and the compressor exit/EXV inlet, as indicated in the System Schematic Diagram. The gas cooler needs to be secured to the chassis, either through the bottom sheet metal or to the fan support strut. The flow arrangement should be counter-flow.

Design and Operational Constraints

In addition to the outdoor and storage conditions, the gas cooler is located on the high-pressure side of the system, and should be designed for both refrigerant and potable water heat exchange:

- Meets refrigerant high-side max operating pressure and min burst pressure
- Refrigerant temperature range: 32 °F to 350 °F.
- Maximum allowable water-side pressure: 120 psig.
- Water temperature range: +33 °F to +210 °F.
- Additional constraint: Double-wall heat exchanger or equivalent for potable application.
- Construction material should compatible with potable water on the hydronic side.

Technical Characteristics

- The maximum capacity of the gas cooler is 25 tons

SERVICE MODULE – EXV *(supplier: Saginomyia)*

The function of the EXV is to expand the refrigerant from high to low pressure. For the CO2 HPWH, it also serves to control high-side pressure for optimal system efficiency and defrost.

Installation and Connections

The EXV connects the gas cooler exit and evaporator inlet, as indicated in the System Schematic Diagram. It is part of the service module.

Design and Operational Constraints

In addition to the outdoor and storage conditions, the EXV is connected to both the low-pressure and high-pressure sides of the system, and is subject to the following constraints:

- Meets refrigerant high-side max operating pressure and min burst pressure.
- Meets appropriate maximum pressure drop conditions
- Meets appropriate refrigerant temperature conditions

SERVICE MODULE – DEFROST VALVE

The function of the defrost valve is to provide hot refrigerant bypass from the compressor exit to the evaporator to provide defrost functionality.

Installation and Connections

The defrost valve connects the compressor exit and EXV inlet, as indicated in the System Schematic Diagram. It is part of the service module.

Design and Operational Constraints

In addition to the outdoor and storage conditions, the defrost valve is connected to the high-pressure side of the system, and is subject to the following constraints:

- Meets refrigerant high-side max operating pressure and min burst pressure.
- Meets appropriate maximum pressure drop conditions
- Meets appropriate refrigerant temperature conditions

SERVICE MODULE – FILTER/DRYER *(suppliers: Danfoss, Parker)*

The function of the filter/dryer is to capture any particulate contaminants in the refrigerant side of the system.

Installation and Connections

The filter/dryer connects the gas cooler outlet to the EXV inlet, as indicated in the System Schematic Diagram. It is part of the service module.

Design and Operational Constraints

In addition to the outdoor and storage conditions, the filter/dryer is connected to the high-pressure side of the system, and is subject to the following constraints:

- Meets refrigerant high-side max operating pressure and min burst pressure.
- Meets appropriate maximum pressure drop conditions
- Meets appropriate refrigerant temperature conditions
- Meets appropriate filtration particle size

SERVICE MODULE – SERVICE VALVES *(suppliers: Mueller Brass, Parker)*

The function of the 2-way service valve is to provide a means for charging/discharging the high-pressure side of the system. The function of the 3-way service valve is to provide a means for charging/discharging the low-pressure side of the system and to provide a means for isolating the high-pressure from the low-pressure side of the system.

Installation and Connections

The 2-way service valve is connected to the EXV inlet, and the 3-way service valve connects the EXV outlet to the evaporator inlet, as indicated in the System Schematic Diagram. Both valves are part of the service module.

Design and Operational Constraints

In addition to the outdoor and storage conditions, the service valves are subject to the refrigerant high-pressure side constraints:

- Provides leak-tight seal
- Meets refrigerant high-side max operating pressure and min burst pressure.
- Maximum refrigerant temperature: +120 °C.

EVAPORATOR *(supplier: ECO)*

The function of the evaporator is to provide heat for the heat pump, through heat exchange of outdoor air with cold refrigerant.

Installation and Connections

The evaporator connects the EXV exit and the accumulator inlet, as indicated in the System Schematic Diagram.

Design and Operational Constraints

In addition to the outdoor and storage conditions, the evaporator is located on the low-pressure side of the system, with the following constraints:

- Meets refrigerant high-side max operating pressure and min burst pressure.
- Meets appropriate maximum pressure drop conditions
- Meets appropriate refrigerant temperature conditions
- Refrigerant tubes should be constructed of copper, with bonded aluminum fins.
- The refrigerant inlet should be equipped with a distributor and feed tubes.

Technical Characteristics

- The maximum capacity of the evaporator is 20 tons

<u>ACCUMULATOR</u> *(supplier: Alfa-Laval)*

The function of accumulator is to store excess refrigerant charge during varying operating conditions. It also serves to protect the compressor from any ingestion of liquid and to return sufficient flow of oil back to the compressor.

Installation and Connections

The accumulator connects the outlet of the evaporator to the inlet of the compressor, as indicated in the System Layout Diagram and the System Schematic Diagram.

Design and Operational Constraints

In addition to the outdoor and storage conditions, the accumulator is subject to the refrigerant low-pressure side constraints:

- Meets refrigerant low-side max operating pressure and min burst pressure.
- Refrigerant temperature range: -40 °F to +115 °F.

<u>FAN MODULE – FAN</u> *(suppliers: Simonin, GE, Leroy Somer)*

The fan module consists of the fan blades, fan motor, and associated support structures. The fan provides airflow for the evaporator.

Installation and Connections

The fan draws air through the evaporator, as indicated in the System Layout Diagram and the System Schematic Diagram.

Design and Operational Constraints

In addition to the outdoor and storage conditions, the fan module must meet the following conditions:

- Must be UL listed/approved.

<u>HYDRONIC MODULE – WATER PUMP & VARIATOR</u> *(suppliers: Salmson, Siemens)*

The water pump provides the necessary flow and head to operate the CO2 heat pump water heater. The variator provides speed control of the pump, to meet customer water outlet temperature setpoints over the entire range of operating conditions.

Installation and Connections

The water pump is a component of the hydronic module. The inlet to the water pump is connected to the inlet water port, and the outlet of the water pump is connected to the gas cooler inlet, as indicated in the System Layout Diagram and the System Schematic Diagram.

Design and Operational Constraints

In addition to the outdoor and storage conditions, the water pump and variator are subject to the following constraints:

- The pump shall be of variable speed drive design and capable of being serviced without disturbing piping connections.
- Maximum working pressure: minimum of 120 psig.
- Water temperature range: +33 °F to +210 °F.
- Construction material should compatible with sanitary/potable water.
- The pump shall operate under 460V (+/- 10%), 3 phase, 60Hz.
- The pump shall include a motor protection sensor

HYDRONIC MODULE – ANCILLARY COMPONENTS *(suppliers: TI)*

The ancillary components of the hydronic module include the analog pump discharge pressure gage, water inlet and outlet thermistors, and low-point drain valve. The thermistors provide information to the control system for various operating conditions. The rest of the components facilitate system serviceability.

Installation and Connections

The hydronic module provides the necessary connection points for the inlet and outlet water supply, as indicated in the System Layout Diagram and the System Schematic Diagram. The inlet water port is connected to pump inlet line, and contains the water inlet thermistor. The pump outlet is connected to the gas cooler inlet, with the analog pump discharge pressure gage. The gas cooler outlet line is connected to the outlet water port, and also contains the low-point drain valve and water outlet thermistor.

Design and Operational Constraints

In addition to the outdoor and storage conditions, the ancillary components of the hydronic module are subject to the following constraints:

- All components designed to a maximum operating pressure of 120 psig.
- Water inlet components: water temperature range of +33 °F up to +210 °F.
- Water outlet components: water temperature range of +33.8 °F to +210 °F.
- Construction material should compatible with sanitary/potable water.

Technical Characteristics

- Pump discharge pressure gage range: 0 to 200 psig.

CONCLUSIONS

The objective of this project was to break down the barriers to North American (NA) market acceptance of high efficiency heat pump water heaters (HPWHs). These barriers were defined to be performance, reliability, and first cost.

Past halocarbon-charged HPWH offerings presented significant challenges. Hot water delivery temperature was limited by pinching at the saturated condensing temperature of the refrigerant. To get an adequately high condensing temperature it was necessary to choose a refrigerant with a high critical pressure. This in turn limited the practical saturated evaporating temperature, thus limiting the lowest ambient temperature at which acceptable performance could be achieved. Within this limited operating envelope, good efficiency can be obtained. These units were typically assembled in small quantities, so price could not be leveraged by large quantity and reliability issues could not be investigated and fully resolved. All of these issues kept market acceptance low and the concept has not been able to reach a critical mass.

In this project we have addressed performance concerns through the use of CO2 as a refrigerant. CO2 has a low critical temperature and can give good performance at ambient temperatures as low as -20 °C. To achieve practical hot water delivery temperatures, the CO2 is compressed to a supercritical condition. This allows practical water delivery temperatures of up to 80 °C. The tradeoffs are that the system pressures are very high (>2000 psig) and the high side pressure must be actively controlled to achieve the best efficiency over the range of operating conditions. These constraints can lead to lower reliability than typical HVAC products.

To demonstrate current state of the art in performance, reliability, and cost we assembled eight CO2 HPWH prototypes to NA specifications. These units were based on, and leveraged by, our parallel development of CO2 HPWHs for the European Union (EU) market. The eight trials were placed in the Northeast, Southeast and Northwest regions to evaluate the effect of different climates, and in a wide variety of applications to evaluate the challenge of different demands.

Performance

Table 15.1 summarizes the representative averaged high performance of the eight field trials under different ambient conditions. The entering and leaving water temperatures were maintained at approximately 10 °C and 60 °C, respectively, in each case.

Unit, phase, location	Air Temperature, °C	Observed COP	Design COP	Deviation
FFU, phase 2, outside	-10	1.8	2.4	- 33%
FFU, phase 2, outside	2	2.2	2.6	- 25%
PGN, phase 2, outside	12	2.8	3.3	- 18%
EHS, phase 2, inside	24	3.3	4.0	- 21%
DCH, phase 1, inside	34	3.6	4.4	- 22%

Table 15.1: Representative performance of the CO2 HPWH field trial units

In all cases the performance of the units is approximately 20-30% below target. This performance deficit was observed in the EU HPWH prototypes and was fully resolved by improving quality control on compressor valve plate flatness. With this same condition imposed we expect that the NA HPWH can meet its performance specification under a wide range of conditions with little additional development.

Reliability

To be a successful product we expect the first year failure rate (FFR) of the CO2 HPWH must be less than 7.5%. The field trials did not demonstrate this level of reliability. Table 15.2 summarizes the operating history of each unit, comparing the normalized operating hours to the expected hours for the application and showing the experienced FFR.

Unit	Compressor Hours	Compressor Hours / Year	Planned Unit Hours / Year	Availability (%)	First year failures
GKN (P1)	754	393	< 1000	44	9
UTRC (P1)	2810	1297	2210	63	10
DCH (P1)	3985	2391	5824	52	8
SCF (P2)	3735	2988	4380	82	5
FFU (P2)	1203	1031	6552	68	2
EHS (P2)	1678	1678	4368	50	8
PGN (P2)	1278	1180	5824	58	3
GHG (P2)	247	270	7280	84	0

Table 15.2: Operational reliability data for the field trial units

The units exhibited a very high FFR, with only one unit not experiencing a hardware failure in the first year. The phase 1 units averaged 9 failures/yr, while the phase 2 units had 3.6 failures/yr. Although this is far from the target, it is important to consider the nature of the failures. Figure 15.1 shows a Pareto diagram of the CO2 HPWH failures. If the Pareto for the Carrier Aquasnap chiller (the basis for the CO2 HPWH) is examined, the top five issues are controls (3 times as large as any other category), leakage, sensors, wiring, and compressor. Although the failure rates are much lower on this developed product, it can be seen that many of the observed CO2 HPWH failures are very similar to historic Carrier products. There is a high probability ty that failures in these 5 categories can be reduced to levels acceptable to a major product manufacturer.

The standout reliability issues are the EXV and the excess number of leaks from pressure relief valves (PRVs) and threaded fittings. Replacing the PRV with a burst disc, a common Carrier practice, and replacing threaded fittings with brazed fittings would reduce the leaks dramatically. On the other hand, the EXV failures are persistent and must be resolved in cooperation with the manufacturer. Root cause analysis performed to date suggests that the current design is inadequate, but also that it is feasible to manufacture a reliable valve. It is notable that UTRC has worked closely with the EXV vendor to develop a qualification test plan and qualify the valves to the extent possible, but unexpected failure modes were still observed.

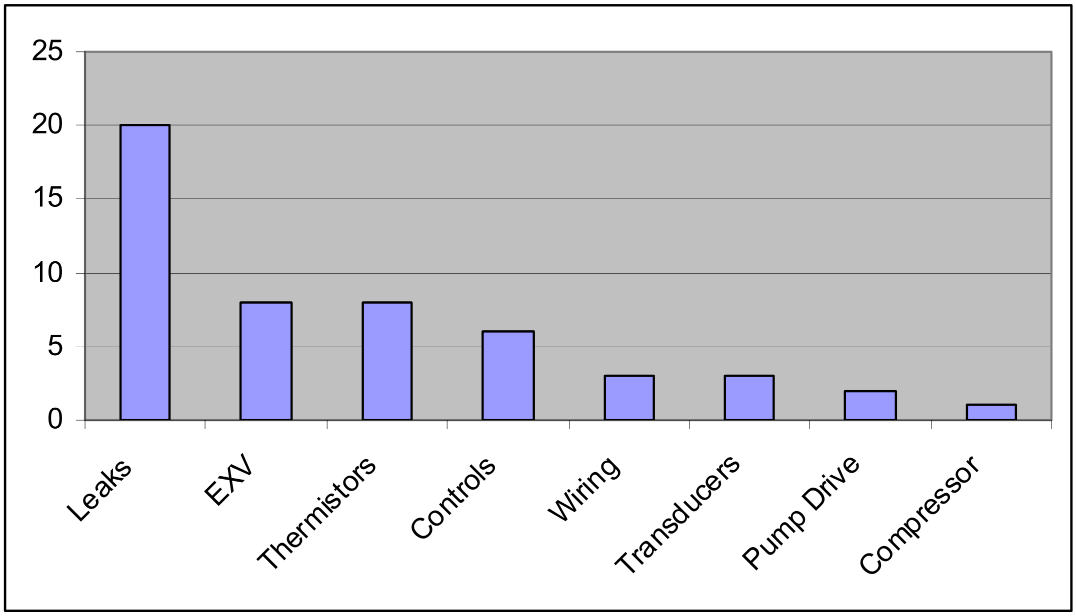


Figure 15.1: Pareto chart of the failures observed in service and at the final examination

The units exhibited a reasonable level of availability, averaging 60% over 11 unit-years. This statistic is misleading in that we were not able to repair alarmed units promptly. Several issues contributed to this. First, communications were unreliable on many units so the notification that the unit was shut down on alarm often came from site personnel. Also, the unexpected number of failures in 2006 taxed the limited resources available for maintenance. We typically waited until several units had shut down before sending a technician to the sites to examine the units and reset or repair them. If two days were allowed for repairing each failure and the discretionary waiting time removed from the availability calculation, the units were capable of averaging near 90% availability.

First cost and market acceptance

At the present state of development we project that first cost at quantity production will be approximately 20% higher than target, although still half that of HPWHs currently on the NA market. The high cost is primarily caused by the need for ancillary equipment to improve the lifetime of the compressor and the cost of the double-walled gas cooler needed for sanitary applications. Installation costs were highly variable, with the best results occurring when the unit was installed by plumbing and electrical contractors rather than HVAC contractors. Given the current cost, and recognizing fluctuations in gas and electricity costs, we predict that displacing an electric water heater with a CO_2 HPWH would result in a payback period between 8 and 24 months. This is adequate for most commercial ventures but is longer than the 6 month payback period market experts claim is necessary for acceptance.

Final analysis

The CO_2 HPWH, in its current state of development, provides efficiency and operating range sufficiently greater than existing products to impact market demand. Although first cost currently results in payback periods 50-400% greater than the 6 month payback currently demanded by the market for the existing HPWH products, higher performance combined with industry brand name recognition may be enough to bridge this payback gap.

Reliability remains the critical issue. Market experts recognize that the incremental cost of hot water is low, but the cost of lack of hot water is high. The commercial market will adopt HPWHs only if they have “drop and forget” levels of reliability, and a brand name commercializer should not risk developing this reliability by launching an underdeveloped product and fixing problems in the field. Our analysis and experience suggests that this level of reliability can be attained from the final unit specification, but that a moderate development and qualification period would be necessary. The reliability tools developed under this project have already been applied to improve the component qualification test plans. Their continued use could help to minimize this development time.

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